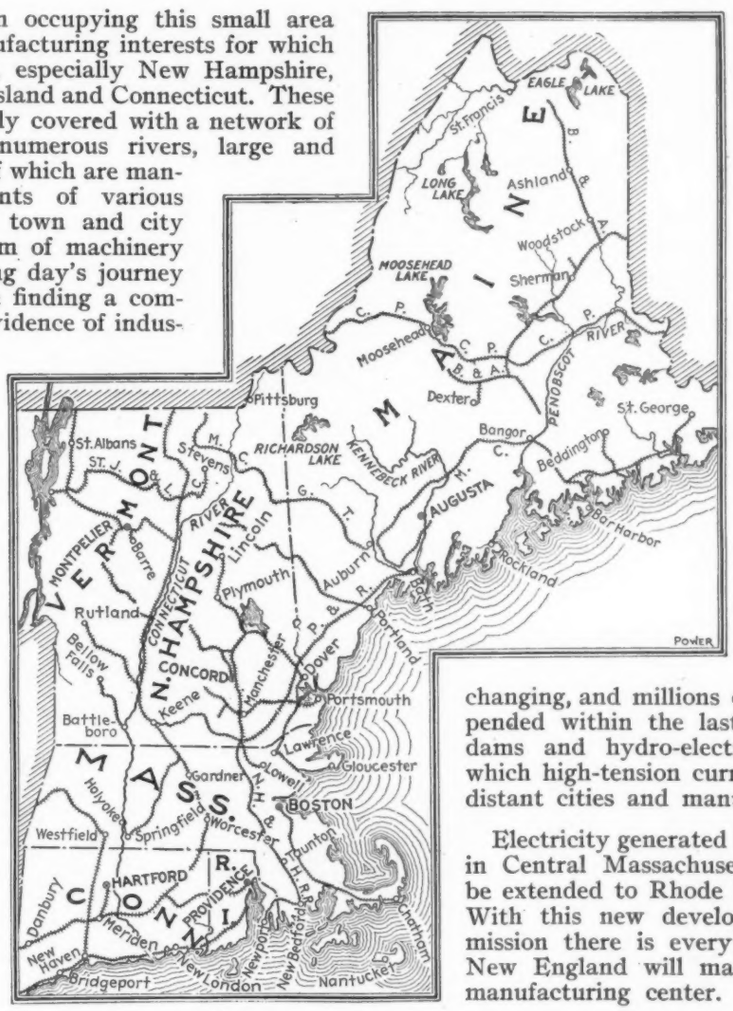
POWER

Vol. 36 NEW YORK, NOVEMBER 26, 1912 No. 22

New England of Today

FEW realize that the six New England states, although having less than one forty-fifth of the area, have about one-fourteenth of the inhabitants of the United States.

This large population occupying this small area is due to the vast manufacturing interests for which New England is noted, especially New Hampshire, Massachusetts, Rhode Island and Connecticut. These four states are practically covered with a network of railroads. There are numerous rivers, large and small, upon the banks of which are manufacturing establishments of various natures. Each hamlet, town and city is enlivened by the hum of machinery and one would go a long day's journey in New England before finding a community free from this evidence of industrial progress.



While the water power of these states plays an important part in the manufacturing field, the steam power plants predominate. The chimneys of thousands of plants stand as evidence to the many industries to which the boilers furnish motive power.

Blowing whistles break the silence of the morning quiet, calling hordes of workmen to their daily toil, and again sound release at night, when the day's work is over, to the same great army of toilers.

Each of these mills necessitates the operation of from one to sixty or more boilers, and a relative engine capacity. Well may it be imagined, then, to what a great number employment is given, of all ranks and degrees of skill from firemen, oilers, etc., to chief engineers, master mechanics, superintendents and consulting and designing engineers.

It would be difficult to estimate the thousands upon thousands of dollars that have been invested in power plant apparatus. These plants range from installations equipped with the simplest apparatus to extensive power plants in which the most economical and modern equipment has been installed.

Until recent years, the output of the electrical power plants of New England was confined to local use, the central stations distributing energy to but comparatively short distances. This condition, however, is changing, and millions of dollars have been expended within the last few years in building dams and hydro-electric power plants from which high-tension current will be delivered to distant cities and manufacturing centers.

Electricity generated in Vermont is being used in Central Massachusetts and eventually will be extended to Rhode Island and Connecticut. With this new development of power transmission there is every reason to believe that New England will maintain its prestige as a manufacturing center.

In this issue will be found descriptions of several power plants in operation in New England, and used for various manufacturing purposes.

Wachusett Dam Hydro Electric Plant

By Warren O. Rogers

The water in the Wachusett Reservoir in flowing to the aqueduct passes through four 1200-hp. turbine waterwheels, each directly connected to a 1000-kw. alternating-current generator. Each waterwheel is capable of passing approximately 44,000,000 gal. of water every 12 hours. The flow is measured by venturi meters.

The generation of electrical energy from water flowing from a reservoir into an aqueduct of a water-works system has been successfully accomplished by the Metropolitan Water Works Board of Massachusetts at the Wachusett dam, Clinton, Mass.

Water supplied to the city of Boston, Mass., by the Metropolitan Water Works is obtained from Lake Cochituate, the Sudbury River and the Nashua River. The former two sources of water supply were acquired by the city of Boston in 1846 and 1872 respectively. The construction work for storing and conveying the water of the Nashua River from the Wachusett Reservoir was begun in 1895 and completed in 1905. Water was first drawn from this source March, 1898.

Water is collected and stored in the Wachusett Reservoir on the Nashua River, in eight storage reservoirs on the watershed of the Sudbury River, and in Lake Cochituate. From these reservoirs the water is conveyed to the Metropolitan District through four aqueducts. The Wachusett Aqueduct, 12 miles long, conveys water from the Wachusett Reservoir to the Sudbury Reservoir. The Western Aqueduct, 12 miles long, conveys water from the Sudbury Reservoir to a point within 10 miles of the Metropolitan District. The Sudbury Aqueduct conveys water 17.4 miles from the reservoir on the Sudbury River to the Chestnut Hill Reservoir, where the well known Chestnut Hill high- and low-service pumping stations, with an aggregate capacity of 204,500,000 gal. for 24 hr., are located. The fourth aqueduct conveys water from Lake Cochituate 13.7 miles to the Chestnut Hill Reservoir, from which the city of Boston and 17 other municipalities obtain their supply.

The Wachusett dam is in a narrow gorge on the Nashua River, about one-half mile upstream from the town of Clinton, Mass. The dam is 1396 ft. long, 207 ft. high and the high-water elevation is 395 ft. above the city of Boston. The area of the reservoir's water surface is 4195 acres and the available capacity is 65,000,000,000 gal. of water. The reservoir is 8.41 miles long with a maximum width of 2.05 miles. The average depth of water is 46 ft. Some interesting figures regarding the dam are given herewith:

Height of top of main dam above full-reservoir level	20 ft.
Height of top of main dam above ground below dam	114 ft.
Maximum height of dam above rock foundation, about	207 ft.
Maximum thickness of dam at bottom, about	185 ft.
Thickness of dam at full-reservoir level	25 ft.
Maximum depth of water above dam	129 ft.
Masonry required for dam	280,000 cu.yd.

Of the 12 miles of the Wachusett aqueduct, two miles is a rock tunnel, seven

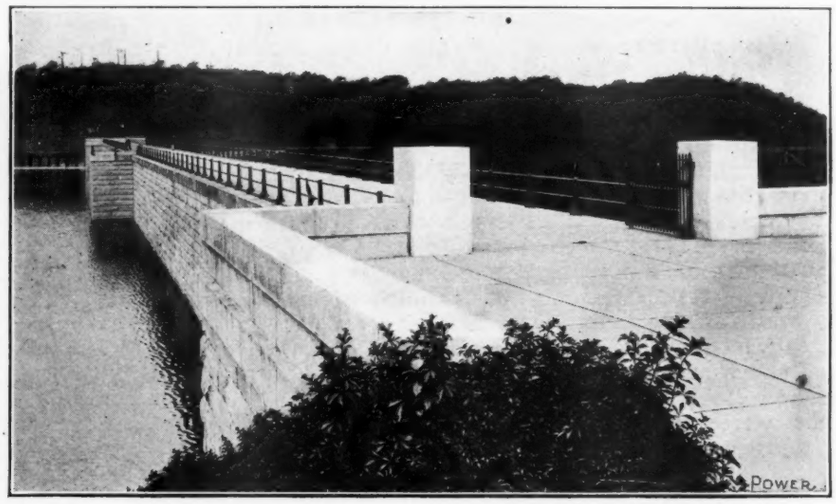


FIG. 2. WACHUSETT DAM PROMENADE

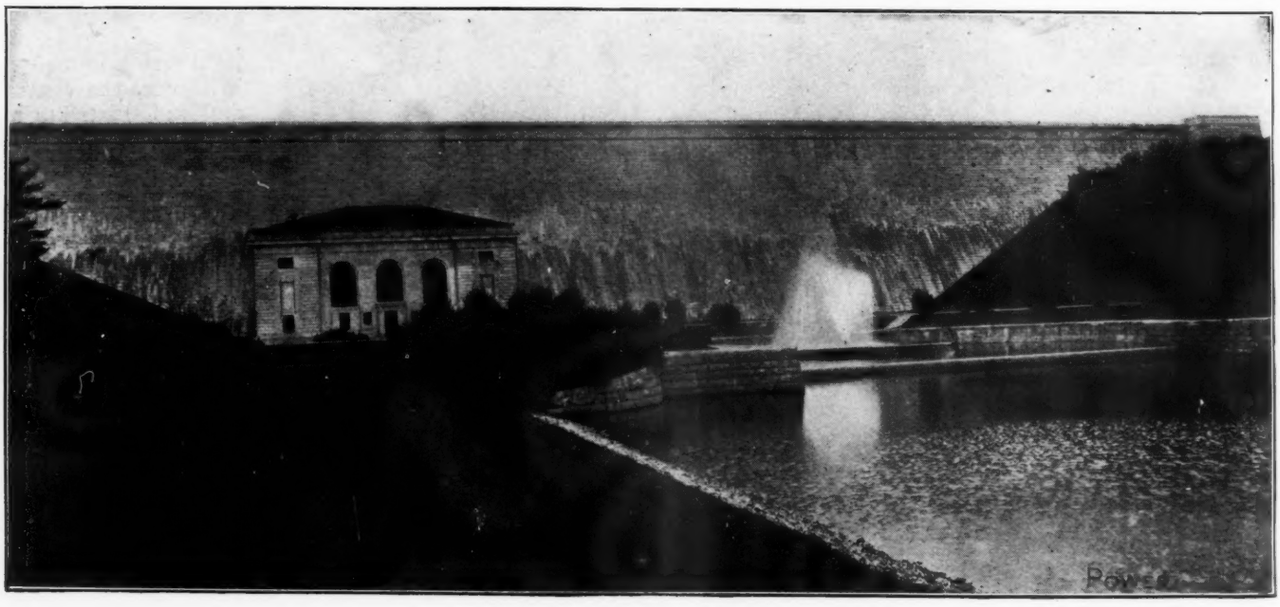


FIG. 1. WACHUSETT DAM AND HYDRO-ELECTRIC POWER HOUSE

miles concrete and brick masonry, covered with earth, and three miles an open channel. The masonry section is 11 ft. 6 in. wide and 10 ft. 6 in. high. The aqueduct has a capacity of 300,000,000 gal. per day.

Although the hydro-electric power-plant building was erected at the time the dam was built, it was not equipped with machinery until recently. It was estimated that about 3000 hp. could be developed by placing turbine waterwheels between the reservoir and aqueduct and utilizing the flow of water passing through them. A view of the dam and power house is shown in Fig. 1. A view of the promenade above the dam is shown in Fig. 2.

There are four 1200-hp. turbine waterwheels of the volute-casing type, each directly connected to a 1000-kw., three-phase, 60-cycle, 13,800-volt, alternating-current generator running at 400 r.p.m. and controlled by a hydraulic governor.

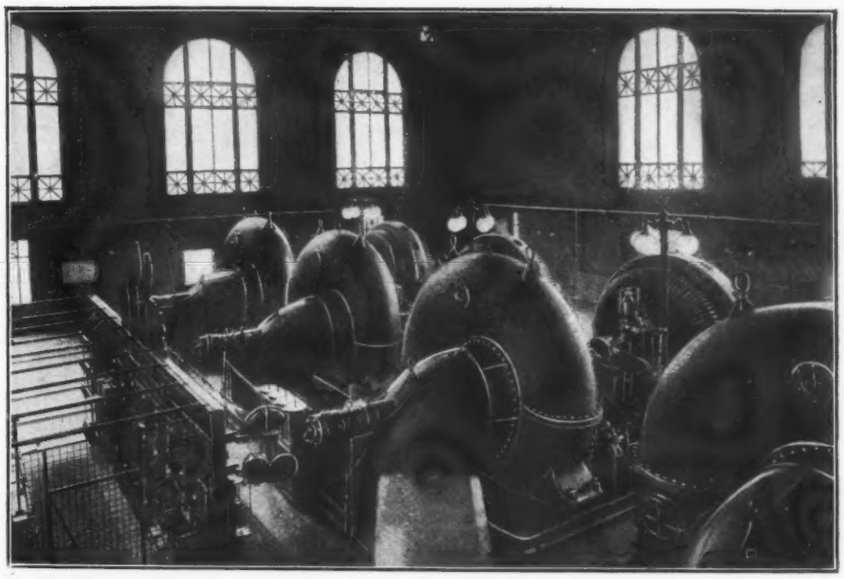


FIG. 4. VIEW OF THE WATERWHEELS, GENERATORS AND SWITCHBOARD

FIG. 3. GENERAL VIEW OF THE ALTERNATING-CURRENT GENERATORS

The units are arranged as shown in Figs. 3 and 4.

Two 60-kw., 125-volt direct-current exciter units are driven by turbine waterwheels, at 1000 r.p.m. They are set in a bay at the end of the generator room as shown in Fig. 3. Details of the installation are given in Fig. 5.

All the water from the Wachusett Reservoir goes through four vertical outlet wells, each 84 in. in diameter and 111 ft. high, and built in the dam. These are connected to 48 in. horizontal cast-iron pipes about 130 ft. long, which are also embedded in the dam masonry, and terminate in the power-house foundation. The turbine wheel foundations form the roof of the aqueduct. After the water passes through the waterwheels it discharges into the aqueduct. A general idea of the dam design and intake-pipe construction may be obtained from Fig. 6.

That accurate measurement of the total amount of water delivered to the aqueduct and also to the turbines may be had, four special venturi type meters have been installed. The meters also act as a check upon the economical operation of the turbines.

It was impracticable to place the upstream pressure chambers in the 48-in. pipe on account of the velocity of the water, about 14 ft. per sec.; therefore a special form of pressure chamber with numerous vent holes, flush with the inner contour of the 84-in. wells was constructed. Special throat pieces 40 in. in diameter are attached to the interior of the 48-in. castings at points approximately 90 ft. from the upstream chambers. These castings have annular pressure chambers and are lined with bronze, bored to the standard venturi curve.

Brass pipes, attached to the interior surfaces of the wells and of the 48-in.

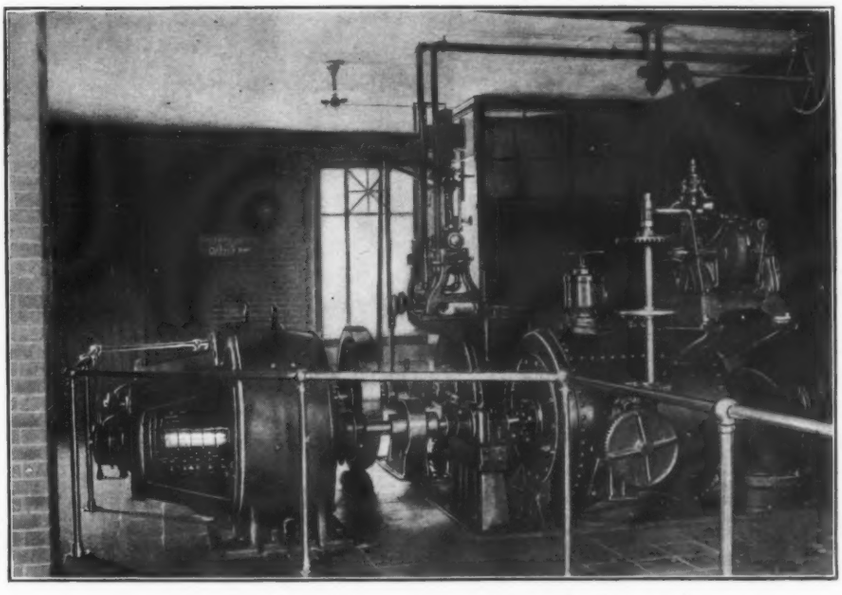


FIG. 5. WATER TURBINE-DRIVEN EXCITER UNITS

pipes, connect with the eight chambers and transmit pressure to the four meters on the main floor of the turbine room. One of the instruments is shown in Fig. 7. Each meter has three dials; one indicates the flow of water in gallons per day, one continually records the rate of flow upon a paper chart and another registers the total quantity of water in gallons.

These instruments are very sensitive, responding to the slightest change in the gate opening. The recording chart, besides registering the amount of water passing through the waterwheels, also shows at what time each started or shut down, and the rate at which it has operated.

At the time of the writer's visit, only two of the four units were running. Through these approximately 88,000,000 gal. of water passed every 12 hours. The water comes to the wheels under a 90-ft. head at a pressure of 37½ lb., there being a slight loss due to friction. The wheels run with a vacuum of 14 lb. Only three of the four units can be run at the same time, as the discharge from them practically equals the capacity of the aqueduct. If it is desirable to shut down all of the machines the water can be bypassed from the reservoir to the aqueduct.

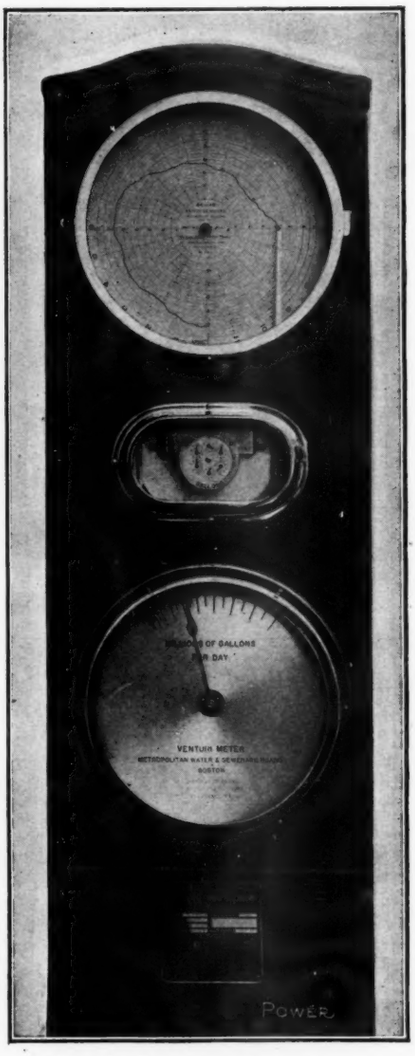


FIG. 7. VENTURI METER BOARD

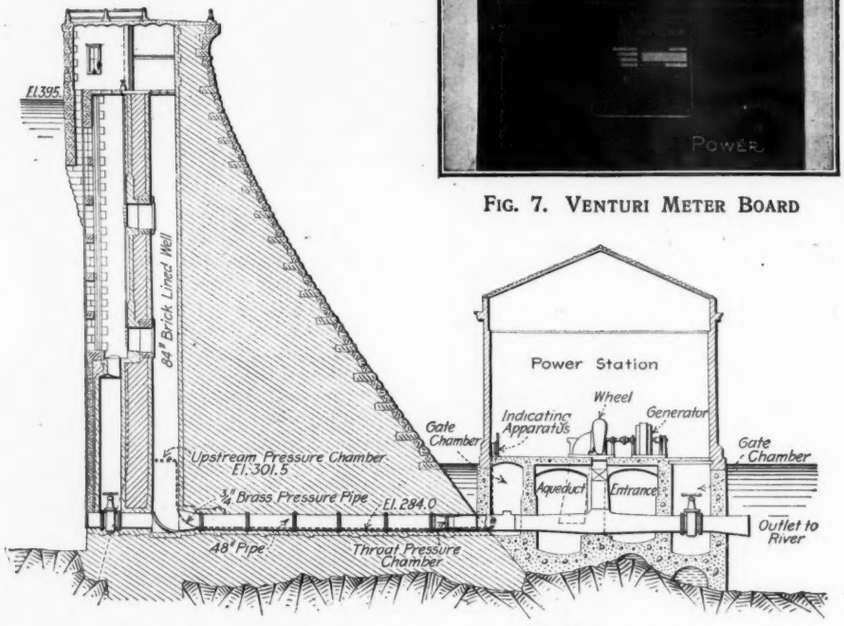


FIG. 6. SECTION OF DAM AND CONNECTION OF VENTURI METERS

The power house is of granite with red-tile flooring. The inside walls are of brick and the ceiling is of oiled hardwood finish. The electrical energy generated at this station is sold to the Connecticut River Power Co., and is transformed by it for commercial use.

Power Cost in a 6500-kw. Station

Figures for the Fall River (Mass.) Electric Light Co.'s plant for the year ended June 30, 1912, are given in *Electrical World*, Sept. 28. The station is a tidewater plant of 6500-kw. rating, containing six 350-hp. water-tube boilers and one 4000-kw., one 500-kw. and two 2000-kw. turbines. The 4000-kw. unit was new in 1911. In 1911 the plant produced 5,764,466 kw.-hr. at a manufacturing cost of $51,840, or 0.9c. per kw.-hr., coal costing $3.67 per ton. The fuel consumption averaged 3 lb. per kw.-hr. This year the plant generated 7,293,783 kw.-hr. at a station cost of $55,359, or 0.76c. per kw.-hr., coal costing the company $3.53 per ton. The payroll of the generating department listed 16 men in 1911 and 18 in 1912.

FALL RIVER STATION OPERATING COSTS
1912

Fuel (0.47c. per kw.hr.)	$34,164
Oil and waste	900
Water	1,206
Wages at station (0.19c. per kw-hr.)	13,687
Repairs of building	1,688
Repairs of steam equipment	1,538
Repairs of electrical equipment	686
Miscellaneous	1,490
Total, excluding fixed costs (0.76c. per kw-hr.)	$55,359

The costs given include no fixed charges or administrative expenses, but show the results of station operation in 1912. The station load-factor for the year was 32.5 per cent.

In recounting his experience in the early days of railroading, when ball-and-lever safety valves were the only kind in use, a veteran locomotive engineer said: "The safety valves were made to blow at 110 or 120 lb. pressure and the amount of steam in the boilers was gaged by the weight required to keep the valve down while the engine was rounding a curve." In those days a boiler would blow itself every time the engine rounded a curve.

EQUIPMENT OF WACHUSETT DAM POWER PLANT

No.	Equipment	Kind	Application	Kw.	Volts	Amp.	Phase	Cycles	R.p.m.	Hp.	Type	Manufacturers
4	Turbines	Water	Main unit	..	....	...	...	....	400	1200	..	S. Morgan Smith Co.
4	Generators	Alt. current	Main unit	1000	13,800	42	3	60	400	..	..	Westinghouse Electric & Mfg. Co.
2	Turbines	Water	Exciter	..	....	...	...	....	1000	..	..	S. Morgan Smith Co.
2	Generators	Direct current	Exciter	60	125	480	...	....	1000	..	..	Westinghouse Electric & Mfg. Co.
4	Governors	Hydraulic	Water wheels	..	....	...	...	....	..	..	Q	Lombard Governor Co.
2	Governors	Hydraulic	Water wheels	..	....	...	...	....	..	..	..	Lombard Governor Co.
4	Meters	Venturi	Water flow	..	....	...	...	....	..	..	..	Builders Iron Foundry
1	Switchboard	Slate	Electrical	..	...	...	...	....	..	..	..	Westinghouse Electric & Mfg. Co.
1	Crane	12 ton	.....	..	....	...	...	....	..	..	..	Niles-Bement-Pond Co.

Power Plant of the Charlton Mills

By Warren O. Rogers

There are six 250-hp. return-tubular boilers with dutch-oven furnaces. A 1350-hp. cross-compound engine exhausts into a 600-kw. low-pressure turbine. A 300-kw. alternating generator, belt driven, is run in parallel with the turbine unit, and acts as a balancer of the electrical load.

Although the generating units in the power plant of the Charlton Mills, Fall River, Mass., are of standard design the arrangement is different from that found in other power plants used for similar work. The mill is adjacent to Cook's pond from which boiler-feed water and condenser-circulating water are obtained.

Boiler Room

The main feature of interest in the boiler room is the application of dutch-oven furnaces to the six 84-in., 250-hp. return-tubular boilers. Five of the furnaces are shown in Fig. 1. These have shaking grates and are hand-fired. When the regular delivery of coal fails to keep up the supply in front of the boiler, coal is wheeled from the pile in the yard.

Instead of being mounted close to the boiler fronts, as in general practice, the water columns and steam gages are brought out to the fronts of the furnaces. The steam and water connections to the water column are supported by hooks attached to vertical I-columns, as shown. A steam gage is piped to the steam connection of each water column by a U-tube and supported on the I-column by a bracket.

Smoke uptakes are placed vertically between the boilers and the main smoke flue. The latter runs above and across the front end of the boilers to a 150-ft. brick chimney, not shown. An economizer heats the feed water from 210 deg. F., as it comes from the exhaust-steam heater, to 270 deg. F., at which temperature it goes to the boilers. Exhaust steam from the auxiliary pumps is used in the feed-water heater.

A main feed pipe runs in front of the furnaces with a branch pipe connected to each boiler. Each branch feed pipe has a gate and globe valve with a check valve between. This permits of examining the check valve at any time.

The boiler house is built of stone and is separated from the engine room by a driveway. The plank roof is covered with tar and gravel.

Engine Room

A view in the engine room is given in Fig. 2. The main unit is a 26 and 42 by 60-in. cross-compound Corliss engine of 1350 hp. capacity, running at 85 r.p.m., with 145-lb. steam pressure. It is belt-connected to the mill lineshaft, which also drives a belted jackshaft in the basement; the latter is used to drive a 300-kw., three-phase, 60-cycle, belt-driven alternating-current generator at 514 r.p.m.

Exhaust steam from the low-pressure cylinder in going to the condenser, passes through a 600-kw. exhaust-steam turbine, Fig. 3. This machine is directly connected to a 750-kw., 600-volt, three-phase, 60-cycle alternating-current generator running at 3600 r.p.m. Exhaust steam comes to the turbine at 7-lb. pressure absolute, and a vacuum of 28 in. is maintained by the condenser to which the turbine is connected. The generating units are in a room on the first floor of the mill building and are supported by brick foundations in the basement.

Exciter Units

Two exciter units generate exciting current for the alternators. One set is a horizontal steam turbine directly driving a 50-kw., 125-volt direct-current generator, running at 2250 r.p.m. The other unit is a 25-kw., 125-volt generator, directly driven by a 32-hp. induction motor at 1150 r.p.m. The motor-driven unit is generally used; the turbine driven one is for use at the noon hour, or during overtime work, when there is a small motor load.

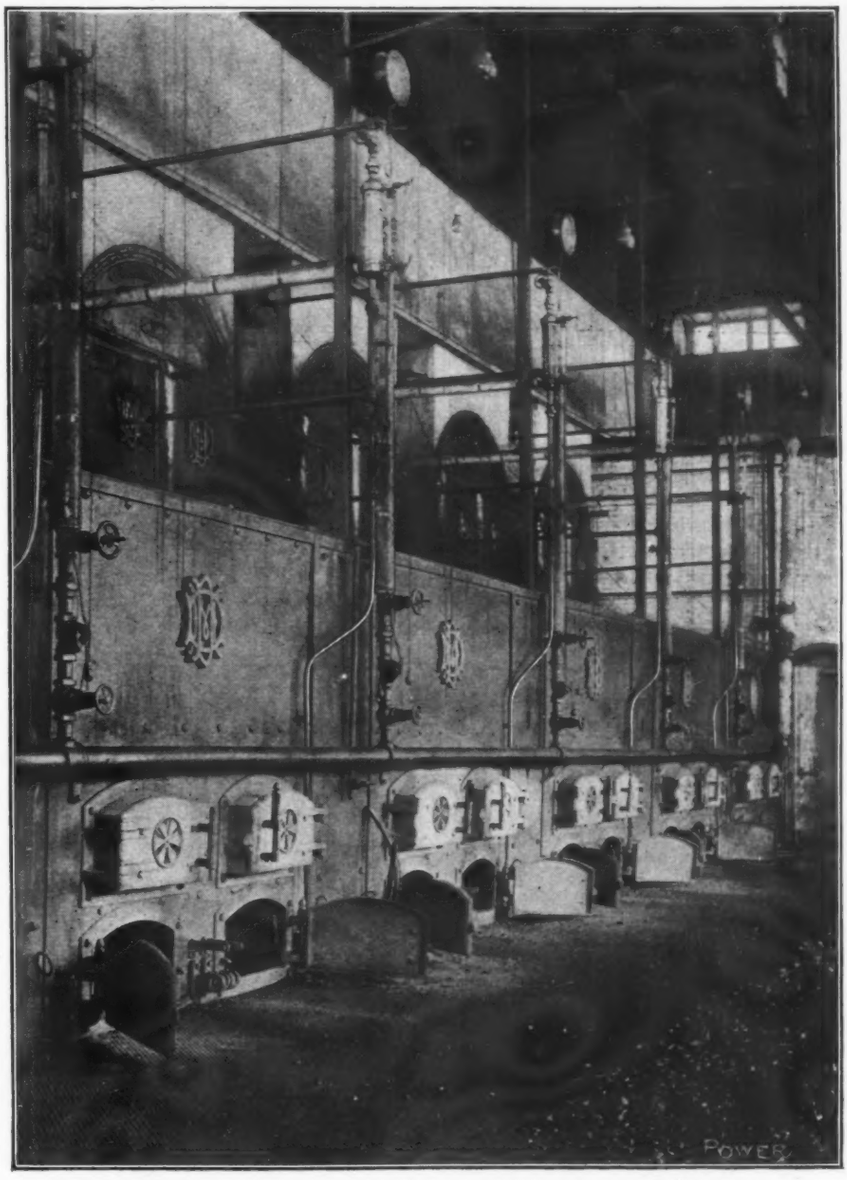


Fig. 1. Return Tubular Boilers and Dutch-oven Furnaces

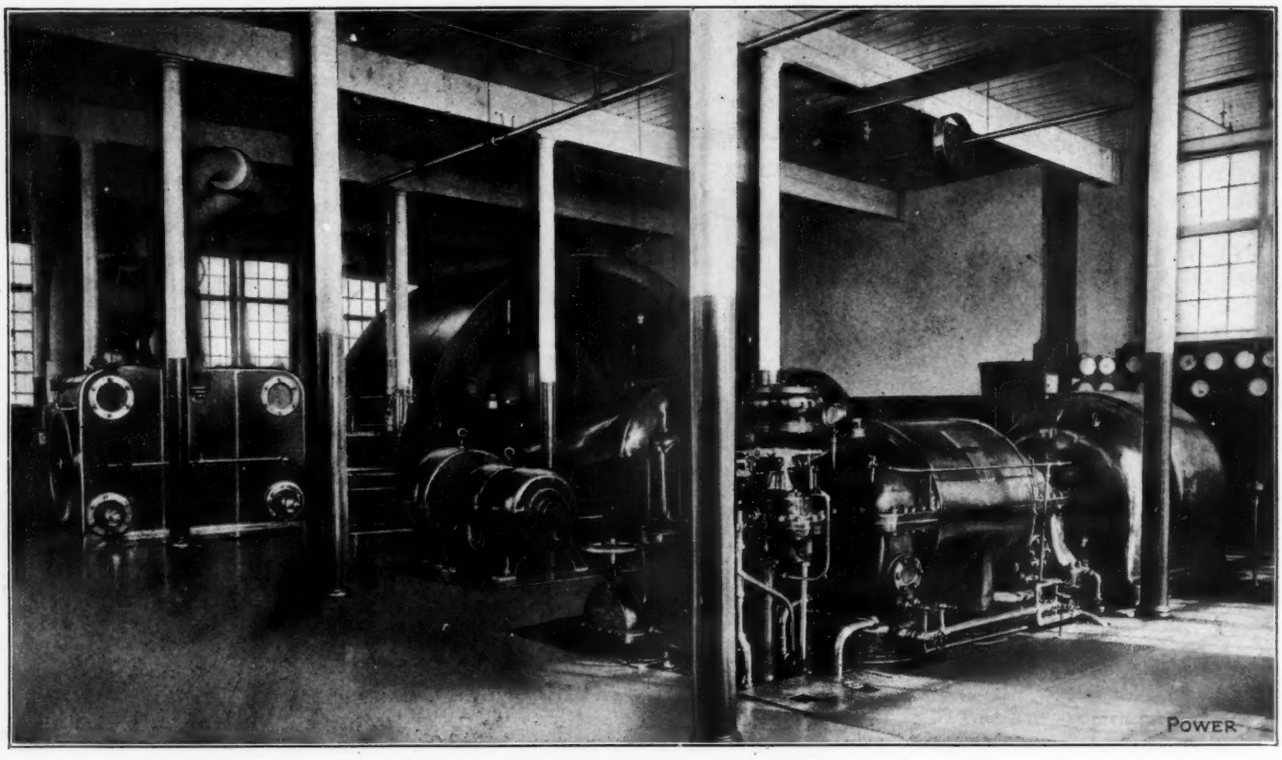


FIG. 2. GENERAL VIEW OF THE ENGINE ROOM

AUXILIARIES

All auxiliaries are in the basement. The condenser is shown in the foreground of Fig. 4. The other equipment includes an 8x10-in. triple-plunger power pump used for pumping water for the humidifying system in the mill, and a 5¼x6¾-in. triple-plunger pump that supplies the mill with water for various uses. For boiler feeding a 5¼x6¾-in. six-plunger power pump is used. These pumps are belt-driven from a shaft hung from the basement ceiling and driven from the jack-shaft in the basement.

A 4½x4½-in. belt-driven air compressor furnishes air for the humidifying system. An 18 and 10 by 12-in. duplex pump is connected to the fire-service lines with which the mill is protected. The exhaust steam from the exciter turbine, condenser turbine and the steam pumps piped to the mill-heating system, goes to the feed-water heater. The returns from the mill and drips from the steam pipe run into a tank from which they are pumped to the boilers. The condenser takes water from a well, under the basement floor, that is connected with the adjacent pond. No attempt is made

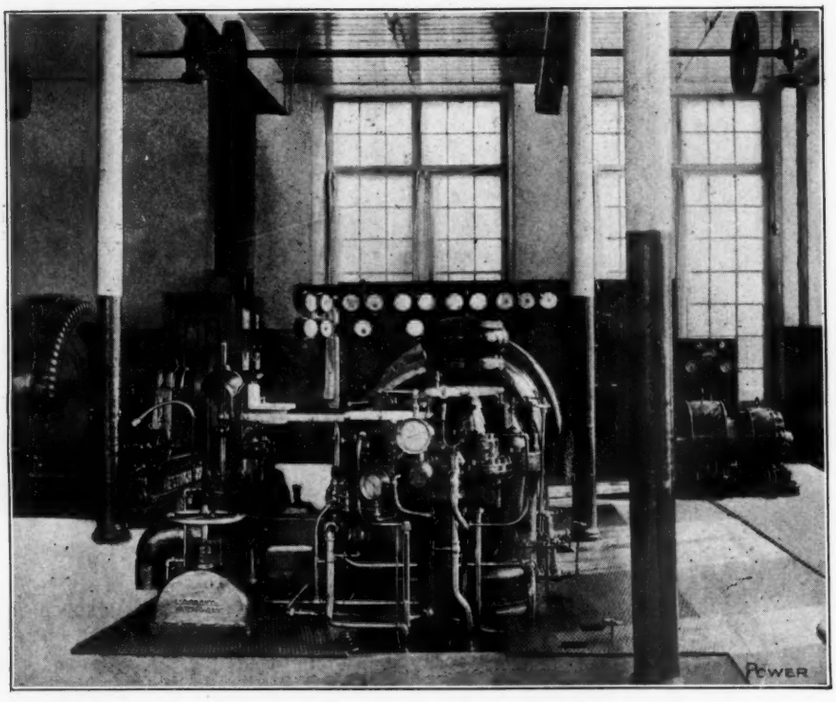


FIG. 3. EXHAUST STEAM TURBINE, MOTOR-DRIVEN EXCITER AND BELT-DRIVEN ALTERNATOR

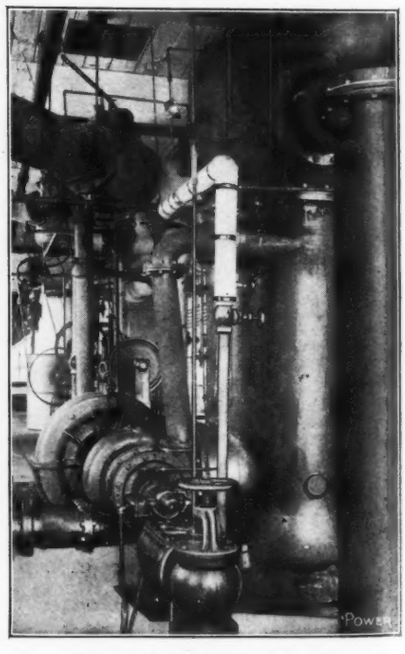


FIG. 4. LEBLANC CONDENSER AND AUXILIARY PUMPS

to save the condensing water and it is discharged to the sewer.

Operation of Alternators

In the mill are 11 induction motors, supplied with current by the alternators. Two direct-current motors in the card room and one in the machine shop are driven by the small direct-current turbine unit. Four of the induction motors are of 100 hp. capacity; the others of 35 hp. capacity. During the noon hour, nights and at other times when it is necessary to run the card room or machine shop, the steam-driven exciter is used. The engine, belt-driven exciter and low-pressure turbine unit are shut down at noon, but the turbine-driven exciter is operated to furnish exciting current for the belt-driven alternator.

After a shutdown the motor-driven exciter, which is left switched in on the switchboard, starts as soon as the alternator generates current, and as it reaches normal speed the exciting load for both alternators is switched to it.

If the steam-exciter unit becomes disabled the motor-driven machine can be belt-driven by a countershaft, shown in Fig. 3, belted to a pulley on the shaft of the belt-driven alternator until the exciter unit is up to speed and the motor switches thrown in, when the belts are removed.

An interesting feature of the plant is the way the two alternators work together. The belt-driven unit runs only when the steam engine is operated. The exhaust-steam turbine is also dependent upon the engine for steam. Upon shutting down, the turbine generator is cut out of circuit at the switchboard and only phased into circuit after both alternators have gained their normal voltage after starting up.

This arrangement of alternators produces a balancing of the load between the engine and turbine-driven units. The load on the alternators is greater than the capacity of the turbine, which tends to slow down the turbine, but as the two alternators are run in parallel, the excess load is thrown on the engine-driven unit. This causes the engine to cut off steam later in the stroke and thus exhausts more steam to the turbine, which again carries its load. This balancing of the load takes place so long as the units are operated in parallel.

A six-panel switchboard carries the recording instruments and controlling equipment for the energy generated by the electrical units. The engine and boiler rooms are light and roomy. The engine room is finished with an oiled, hard-wood floor, pea-green walls with dark-red wainscoting and white ceiling. The plant is under the direct charge of Francis Clegg, chief engineer.

EQUIPMENT OF THE CHARLTON MILLS POWER PLANT

No.	Equipment	Kind	Application	Kw.	Volts	Amp.	Phase	Type	Cycles	R.p.m.	Size	Pres. lb.	Hp.	Manufacturers
6	Boilers	Return tubular	Steam	...	...	..	...	...	..	..	84″x20′	145	250	D. M. Dillon Steam Boiler Works
1	Economizer	Sturtevant	Feed water	...	...	..	...	...	..	..	....	...	...	B. F. Sturtevant Co.
1	Heater	National	Feed water	...	...	..	...	...	..	..	....	...	...	The National Pipe Bending Co.
1	Engine	Rice & Sargent	Main unit	...	...	..	...	...	..	85	26x42x60″	145	1350	Providence Engineering Works
1	Generator	Alt. current	Belt driven unit	300	600	288	3	...	60	514	....	...	...	Westinghouse Electric & Mfg. Co.
1	Turbine	Low pressure	Main unit	600	...	...	...	...	..	3600	....	7 abs	...	Westinghouse Machine Co.
1	Generator	Alt. current	Turbine unit	750	600	725	3	...	60	3600	....	...	...	Westinghouse Electric & Mfg. Co.
1	Condenser	Leblanc	Turbine	...	...	..	...	...	..	...	11	...	..	Westinghouse Mfg. Co.
1	Turbine	Single stage	Exciter unit	50	...	...	...	...	..	2250	....	145	...	Westinghouse Mfg. Co.
1	Generator	Direct current	Exciter unit	50	125	400	...	...	..	2250	....	...	...	Westinghouse Electric & Mfg. Co.
1	Motor	Induction	Exciter unit	...	600	34.2	3	C.C.L.	60	1150	....	...	32	Westinghouse Electric & Mfg. Co.
1	Generator	Direct current	Exciter unit	25	125	200	...	...	..	1150	....	...	...	Westinghouse Electric & Mfg. Co.
1	Pump	Triple plunger	Humidifying water	...	...	..	...	...	..	..	8x10″	...	...	Deane Steam Pump Co.
1	Pump	Triple plunger	Mill water	...	...	..	...	...	..	..	5½x6½″	...	...	I. B. Davis
1	Pump	Six plunger	Boiler feed	...	...	..	...	...	..	..	5½x6½″	...	...	I. B. Davis
1	Compressor	Belt driven	Air	...	...	..	...	...	..	..	4½x4½″	...	...	Ingersoll-Rand Co.
1	Pump	Duplex	Fire	...	...	..	...	...	..	..	18x10x12″	...	...	Warren Steam Pump Co.
1	Switchboard	Slate	Electrical control	...	...	..	...	...	..	..	....	...	...	Westinghouse Electric & Mfg. Co.

A Power Plant Data Sheet

By J. C. Hawkins

The accompanying data sheet was laid out for an 800-hp. power plant. It can be modified to suit the requirements of any ordinary demand.

Power-plant records are necessary in keeping track of the efficiency of the plant. Before an engineer can calculate the cost of the energy generated he must have the data at hand from which to make his calculations.

The systems of records used in most large central stations are necessarily complicated and each item is figured out in detail. This is desirable, but the clerical work attached thereto prohibits its use in the small plant, where the engineer has to do his own bookkeeping, but every steam plant should have some form of daily record of its operation. After several years' experience in plants where various kinds of records were kept, the one shown herewith was drafted for present use, but may be changed to fit any conditions.

The sheet is 13x16 in. and is filed in a loose-leaf file. To illustrate the scope of the record sheet and make it more easily understood, a plant of 800 hp. capacity is taken as an example, in which are four engines and four boilers, a heating system, using forced circulating hot water, the heaters using exhaust steam, supplanted by live steam when necessary, and also furnishing hot-water service. There is also an ice-making plant and a vacuum-cleaning machine, the general layout being such as is found in a modern office building. The sheet is laid out for three eight-hour watches.

Under the heading "Data" the various temperatures and pressures are recorded every two hours, or oftener if required. The outside temperature and condition of the weather will help greatly in keeping the water in the heating system at the proper temperature; any change in the outside temperature is met by a change in the heater temperature before that of the building varies to any extent.

The temperature of the brine in the ice tank is recorded, as any change will affect the output of ice. It is also well to take the temperature of the gas in the suction and discharge pipes and that of the cooling water. The four items, temperature of flue gas, per cent. of CO_2 in the flue gas, draft and condition and time of cleaning the fires, play a very important part in the economical operation of the plant, for it is here that much of the waste of the plant occurs and even if suspected is apt to be neglected.

To obtain this information, a high-reading thermometer or pyrometer, a draft gage and some form of CO_2 testing out-

fit will be required. If a CO_2 recorder is used, its reading should be noted so that the temperature, draft, etc., can be readily compared. If the fires are given a reasonable amount of attention, the use of these instruments will soon pay for themselves in the increased economy of the boilers.

Both volt and ampere readings should be noted, as they are a check on the operation of the boilers and engine, also the variation of load during the 24 hr. This data is noted on the sheet every two hours, or oftener, if desired, by the assistant engineer or oiler. An examination of the report will show any variation in the unit of output per unit of input, and after the engineer has watched this data for awhile, any variation in the various items will show just where the trouble lies.

Weights of coal, ashes and feed water should be recorded at the end of each watch, as they will not only reveal any mismanagement on the part of the employees, but will show just what conditions of fire draft, per cent. CO_2, etc., will give the highest evaporation per pound of coal, and whether or not instructions are being carried out.

Where live steam is used to help out the exhaust steam in heating or for any purpose of manufacture, it should be metered, or the condensation measured if more convenient. If used in heating, any increase in live steam used in proportion to the temperature changes, condition of the weather, exhaust steam, etc., would indicate waste somewhere, and would lead to an investigation; also measuring the live steam and deducting it from the total weight of steam made, and dividing this by the horsepower will show whether the steam consumption of the engine is as low as it should be, and help to locate any unseen steam leaks or other waste.

Under the heading "Operation," the time of starting and stopping each piece of machinery is recorded, also the hours run. This is desirable for many reasons, one of which, in the case of the boilers, is the period of service since cleaning. If any piece of machinery is in operation 24 hr. a day without a stop, the engineer on the first watch when making out a new sheet, writes "On" in the column headed "Start" and the engineer finishing the last watch marks up 24 hr. in the "Hours Run" column.

THE W. H. JOHNSON COMPANY
DAILY REPORT OF ENGINEER

191

Data	First Watch 7 A.M.	9 A.M.	11 A.M.	1 P.M.	Second Watch 3 P.M.	5 P.M.	7 P.M.	9 P.M.	Third Watch 11 P.M.	1 A.M.	3 A.M.	5 A.M.
Steam Pressure												
Exhaust Pressure												
Outside Temperature												
Weather												
Temp. B'l'd'g H't'n'g Sup'y												
Temp. B'l'd'g Return												
Temp. H.W. Service												
Temp. Brine in Ice Tank												
Temp. Feed Water												
Temp. Eng. Room												
Temp. Flue Gas												
% CO_2 in Flue Gas												
Draught-Inches												
Fire Cleaned-Condition												
Voltmeter												
Ammeter												
Coal and Ashes	Coal		Ashes		Coal		Ashes		Coal		Ashes	
Feed Water Meter	Read		Used		Read		Used		Read		Used	
Live Steam Meter	Read		Used		Read		Used		Read		Used	

Operation	Start	Stop	H'r's Run		Start	Stop	H'r's Run
Engine No.1				Boiler Feed Pump No.1			
Engine No.2				Boiler Feed Pump No.2			
Engine No.3				General Service Pump			
Engine No.4				Vacuum Pump No.1			
Fan Engine No.1				Vacuum Pump No.2			
Fan Engine No.2				Circulating Pump No.1			
Boiler No.1				Circulating Pump No.2			
Boiler No.2				Sump Pump			
Boiler No.3				Vacuum Cleaning Machine			
Boiler No.4				Ice Machine			
Fire Pump				Coal Crusher & Conveyor			
Assistant Engineer							
Oiler							
Firemen							
Coal Passer							
Other Labor							

Wattmeter	Positive Lt'g	Positive Pow'r	Diff'n'c	Positive Lt'g	Positive Pow'r	Diff'n'c				
Unit No.1							Total Coal		Total K.W.	
Unit No.2							Total Ash		Total L.S't'n	
Unit No.3							Total F.W't'r		Ice Pulled	
Unit No.4							Evap.P'r.L'b'c		Oil, C. & Waste	
Power Load*			K.W's				Coal R'c'd		Ashes Rm'vd	

Remarks, Repairs, Material used, etc. cont'd. otherside.

Chief Engineer

Power

SIMPLE AND COMPACT DATA SHEET

All generators are on the three-wire system with the lighting circuits tapped between the outside and neutral wires, with a wattmeter in each side between the machine and busbars. The power circuit is taken from the two outside busbars, double the lighting voltage. The readings of the wattmeters on the machines give the total kilowatt load and the reading of the power wattmeter subtracted from this gives the total kilowatt lighting load. With several buildings supplied it would be desirable to have a meter on the main circuit to each building.

Under the heading "Totals" the total amount of coal, water, ashes, live steam, etc., are recorded, also oil, grease and waste, and coal received. The wattmeters are read and the totals figured by the engineer at the end of the third watch, and the meter readings are copied on the new sheet by the next man when coming on duty.

Under remarks, repairs, etc., are recorded all repairs and changes made, the material used and any other information that may be of interest, and signed with the initials of the engineer in charge of the work.

After the report is completed it is examined and signed by the chief engineer or superintendent and placed on file for future reference.

Each individual plant will require some change in the items to make the report sheet fit the local conditions, but the general makeup will fit many plants, and enough information may be had from it to keep the plant in first-class condition if use is made of the information that is gained.

Looking for a Substitute for Gasoline

What promises to be an investigation of more than usual interest and utility is being conducted by the State Engineering Experiment Station at the Pennsylvania State College under the direction of Prof. J. A. Moyer, of the mechanical-engineering department. With the increase in the price of gasoline has come the demand for some cheaper fuel that will give as good results. Kerosene would meet the demands if a satisfactory carburetor could be designed, and it is with a view of determining the merits and defects of various types of carburetors that the investigation is being carried on. The price of gasoline has nearly doubled in this country in the past year, while in England it has risen correspondingly. Many advocate the use of a mixture of gasoline and kerosene, but point out that without an efficient carburetor no such mixture may be used. Prof. Moyer has already installed at the college a motor built to utilize the energy from either of the two fuels mentioned.

Steam and Hydro Electric Power Plant

By R. O. Warren

By installing a hydro-electric unit, the ordinary station load is carried without the aid of the steam engines, which are held in reserve.

At peak loads or during stages of low water, the load above the capacity of the turbine is carried by one of the steam units.

For many years the smaller towns and cities of New England have depended upon small steam power plants for local lighting and power. During these years many thousands of water horsepower have been allowed to go to waste, with little or no attempt to utilize the natural resources with which these states abound.

A change is taking place, however, and hydro-electric plants are being installed to utilize this water power which, in many instances, has been considered of no commercial value.

FIG. 1. POWER PLANT AND CANAL AT WINDSOR LOCKS, CONN.

The Northern Connecticut Light & Power Co., Windsor Locks, Conn., can be taken as an example of what is being done in this direction. Its power plant is at the lower end of a canal supplying several mills with water power. It is built of brick, as shown in Fig. 1.

An interior view of the engine room, Fig. 2, shows the layout of the three steam engines. One is a 13 and 25 by 16-in. double flywheel, center-crank, compound engine, belt connected to a 165-kv.-a. generator running at 600 r.p.m. The generator shaft extends through the bearing at one end, upon which two pulleys are mounted and which carry the two belts from the engine. A third pulley carries a belt driving a compound-wound, 11-kw., 125-volt exciter running at 11.25 r.p.m.

A 13 and 25 by 16-in., 325-hp. condensing Corliss engine is belted to a 200-kw. alternator. The engine runs at 150 r.p.m. and the generator at 600 r.p.m. A 120-volt, direct-current exciter-generator is belt driven from the alternator shaft at 1200 r.p.m.

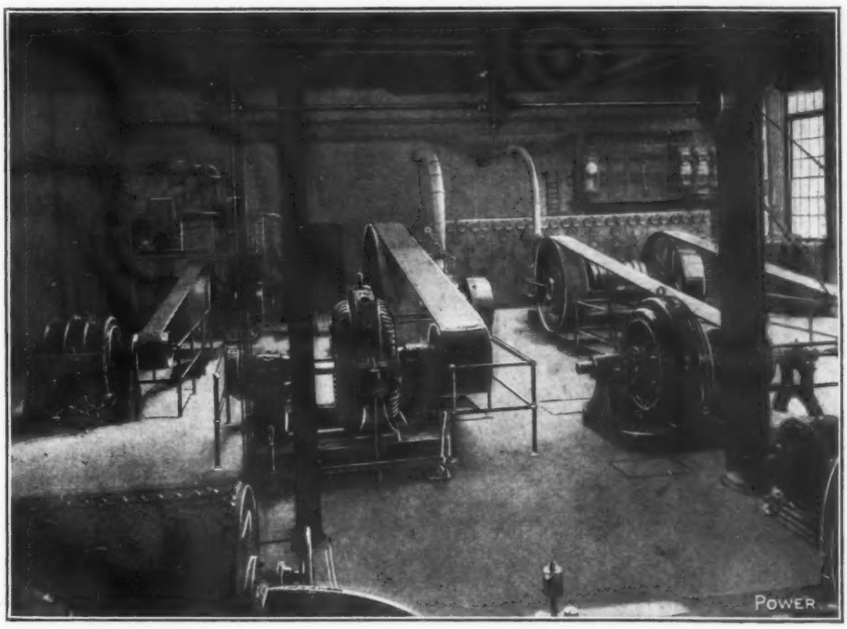


FIG. 2. PARTIAL VIEW OF THE ENGINE ROOM

The third unit is driven by a 12 and 19 by 16-in. vertical compound engine of 250 hp. It is belt connected to a 135-kw. generator running at 900 r.p.m. The 120-volt exciter for this unit is belt driven from a pulley on the generator shaft at 1100 r.p.m.

These units receive steam from three return-tubular boilers, Fig. 3. Forced draft is supplied under the grates by a steam turbo-blower for the limited period the boilers are in service, which is during the peak load each day. The engines run condensing and are piped to two barometric condensers. One has been made from the condenser part of an old jet pump. A vacuum of 25 to 26 in. is maintained.

The day and night load is carried by a 250-hp. water turbine and generator. This unit is at the rear of the engine and is shown in Fig. 4. The wheel runs at 277 r.p.m. and takes water from the canal at a 26-ft. head. The water is discharged through a steel penstock to the Connecticut River. A hydraulic waterwheel governor controls the speed of the unit.

Directly coupled to the shaft of the waterwheel is a 180-kw. generator. A 125-kw., direct-current, 125-volt exciter is belt driven from the generator shaft. This unit is relieved of peak loads by one of the steam-driven generators. Also at periods of low water. The four generators are three-phase, 60-cycle, 2300-volt machines.

A blue Vermont-marble switchboard at one end of the engine room contains four exciter panels, four generator

panels, one transmission panel, one local totalizing panel, three commercial lighting and power panels and one street-lighting panel.

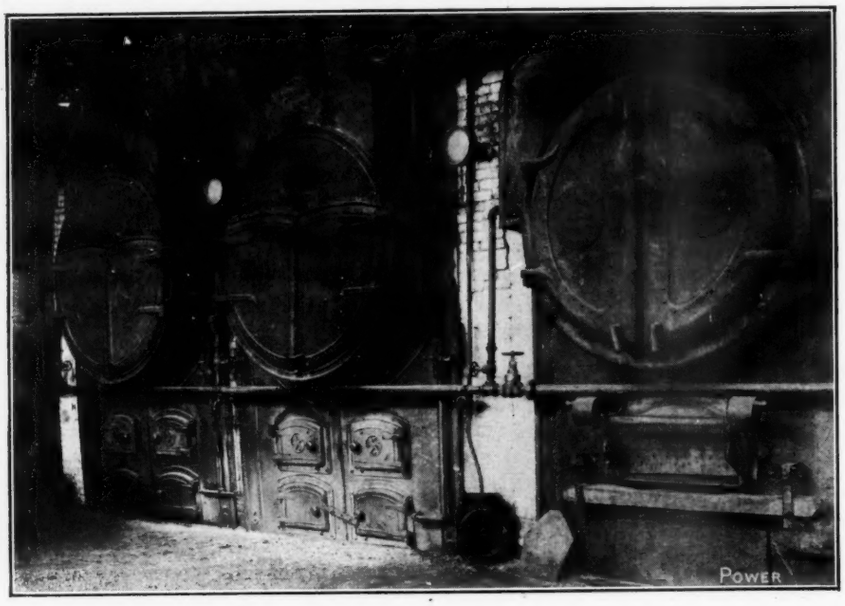


FIG. 3. STEAM BOILERS AND STEAM-DRIVEN BLOWER

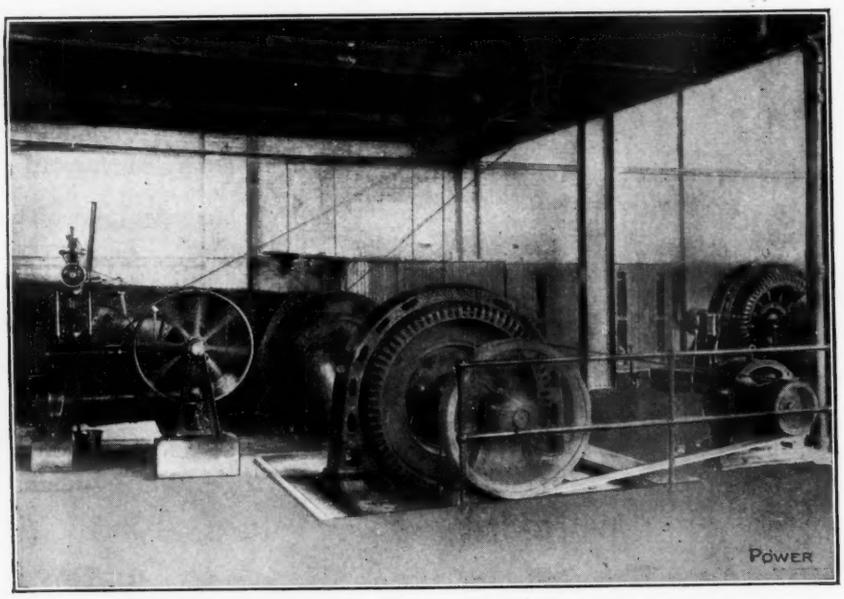


FIG. 4. 250-HP. WATER-TURBINE GENERATING SET AND HYDRAULIC GOVERNOR

Three 75-kw., 2300-6600-volt transformers step up the potential for the circuits to the Thompsonville substation. The longest transmission line is 17 miles. The average load of the station is 1600 kw.-hr. during the summer months and 2700 for the winter months. A record is kept of what the plant is doing, readings being taken at stated intervals and the record filed in the office.

Bearing Data Sheets and Binder

The New Departure Manufacturing Co., Bristol, Conn., has got out a series of data sheets devoted exclusively to the adaptability of "New Departure" ball bearings to machine tools and shaft hangers.

These data sheets cover the three types of bearings manufactured, each of which has its peculiar and particular qualifications. The sheets are in convenient letter size, bound in a loose-leaf binder, and give technical data of specific mountings in milling-machine heads, speed cones and drive pulleys of drilling machines, air-compressor bases, high-speed grinding spindles, grinding or polishing heads, etc.

Other sheets describe the construction of a shaft hanger equipped with a special ball bearing, friction-elimination qualifications under varying loads and a self-aligning feature for this character of work. The sheets and loose-leaf binder will be sent upon request.

What is claimed to be the largest steam turbine-driven centrifugal pump ever built in this country is to be installed by the De Laval Steam Turbine Co., Trenton, N. J., in the Ross pumping station of the Pittsburgh water-works. The rated capacity of the unit is 100,000,000 gal. per day, against a total head of 56 ft., or more than 980 water horsepower. The pump is guaranteed to show a duty of 115,000,000 ft.-lb. per 1000 lb. of dry steam, all the steam used by the condensing equipment being charged against the main unit.

MECHANICAL EQUIPMENT OF THE NORTHERN CONNECTICUT LIGHT AND POWER CO.

No.	Equipment	Kind	Purpose	K.w.	Volts	Amp. per terminal	Phase	Cycles	R.p.m.	Size	Hp.	Pres. lb.	Manufacturers
1	Engine.....	Corliss	Main unit	...	..	...	.	..	150	16x30"	325	120	Allis Chalmers Co.
1	Engine.....	Ball	Main unit	...	..	...	.	..	225	13x25x16"	300	120	Ball Engine Co.
1	Engine.....	Erie	Main unit	...	..	...	.	..	..	12x19x16"	250	120	Lake Erie Engine Co
1	Generator.....	Stanley	Main unit	135	2300	...	3	60	..	..	.	...	Stanley Electric Co.
1	Generator.....	Crocker-Wheeler	Main unit	165	2300	41.1	3	60	600	..	.	...	Crocker-Wheeler Co.
1	Generator.....	Bullock	Main unit	200	2300	44	3	60	600	..	.	...	Allis Chalmers Co.
1	Generator.....	Crocker-Wheeler	Main unit	180	2300	45.2	3	60	277	..	.	...	Crocker-Wheeler Co.
1	Generator.....	Direct current	Exciter	11	125	88	.	..	1250	..	.	...	General Electric Co.
1	Generator.....	Direct current	Exciter	...	120	50	.	..	1200	..	.	...	Allis Chalmers Co.
1	Generator.....	Direct current	Exciter	12	120	16.7	.	..	1100	..	.	...	Crocker-Wheeler Co.
1	Generator.....	Direct current	Exciter	12.5	125	100	.	..	750	..	.	...	Westinghouse Electric & Mfg. Co.
1	Boiler.....	Return tubular	Steam	...	..	...	.	..	..	60"	90	120	Holyoke Steam Boiler Co.
2	Boilers.....	Return tubular	Steam	...	..	...	.	..	..	72"	150	120	D. M. Dillon Steam Boiler Works.
1	Blower.....	Sturtevant	Forced draft	...	..	...	.	..	..	..	..	...	B. F. Sturtevant Co.
1	Water wheel...	Hanover	Main unit	...	..	...	.	..	..	..	250	...	
1	Governor.....	Hydraulic	Water wheel	...	..	...	.	..	..	..	.	...	Lombard Governor Co.
1	Condenser.....	Barometric	Engine	...	..	...	.	..	..	..	.	...	Dean Bros. Steam Pump Works.
1	Condenser.....	Barometric	Engine	...	..	...	.	..	..	..	.	...	Home-made.

Electrical Department

Conducted to be of service to the men in charge of electrical equipment in the power house

An Emergency Rheostat

By C. V. Hull

A short time ago conditions arose which demanded that a 30-hp. 220-volt motor be used with a wide variation in speed and load. The conditions were quite hard to meet because the load would sometimes be quite heavy at low speed and light at high speed.

The range of speed and load conditions made control by changing the field strength of the shunt field entirely out of the question. The compound winding was not used. The motor was required immediately, and there was not sufficient time to get a standard speed-control rheostat. Even if the time had been ample, it would have been unwise to order an expensive rheostat which would not be needed after two or three days. Accordingly, the electrical department arranged the rheostat shown in the illustrations.

The rheostat proper was made up of 2x4-in. hemlock frames, covered with sheet asbestos, and wound about with ½-in. strips of No. 20 galvanized iron, cut from sheets 8 ft. long. These windings of sheet metal were divided into 10 steps. Since it was not practical to use a row of contacts and a sliding lever contact, as is usual, ten 15-amp. double-pole, single-throw switches were substituted in place of single-pole, single-throw switches as none were in stock.

All the switches were opened before the motor was started. Then switch No. 1 was closed and the motor speeded up by closing as many of the small switches as necessary to obtain the desired speed, which could be reduced at will by opening the required number of small switches.

In operation there was so much resistance in the rheostat that the motor would not start, with even a small load, until one of the small switches was closed. It will be seen that the speed of the motor could be controlled by the number of small switches closed or open. Thus it was possible to get almost any speed at any load within the limit of the motor's power.

One voltmeter *B* gave the line voltage while the other *C* gave the armature drop. An ammeter *A* and a wattmeter *D* were connected for collecting the data of the tests.

As a temporary outfit, the rheostat was more than satisfactory. The speed control was simple and accurate and the flat windings radiated heat so rapidly that there was no trouble on that score. The addition of a field rheostat would have given an almost perfect speed control.

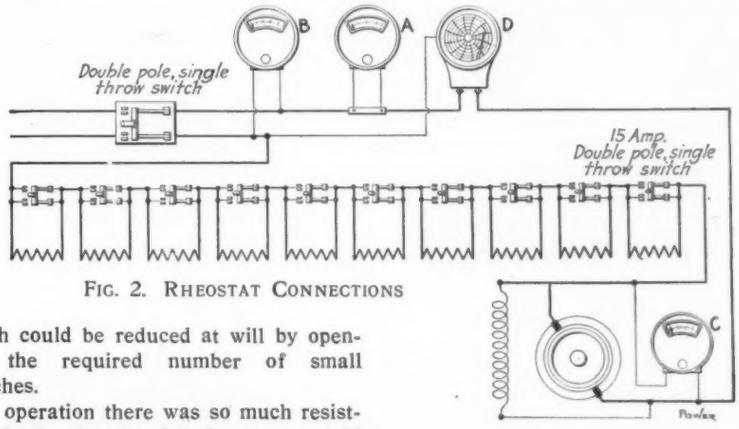


Fig. 2. Rheostat Connections

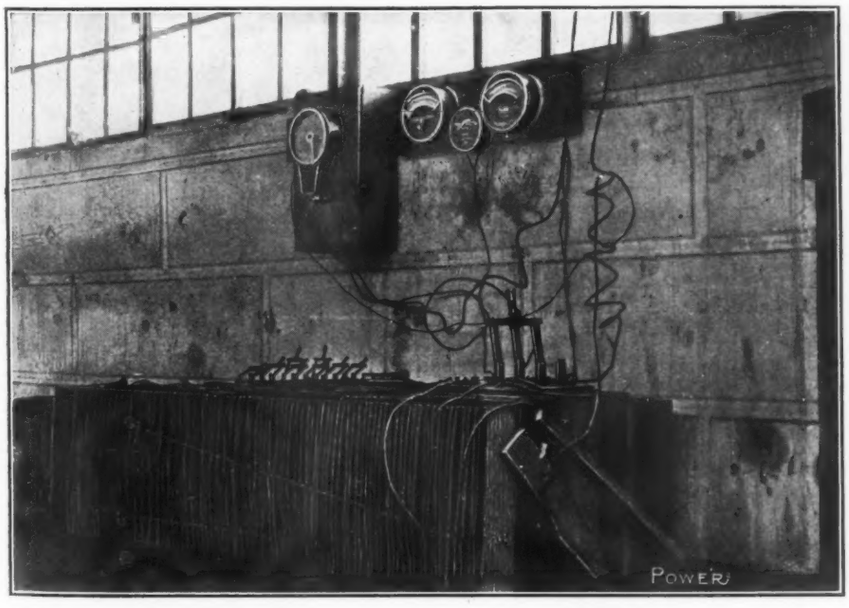


Fig. 1. View of Rheostat and Instruments

Simple Synchronizing Method

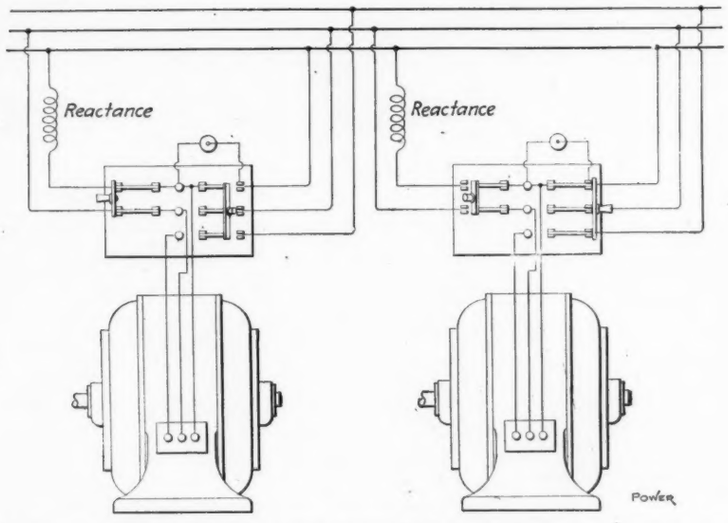
By Gordon Fox

The operation of synchronizing alternators in large central stations has long been accomplished with comparative ease. In the larger plants there are two or more men on duty and it is possible for one to manipulate the throttle or governor of the incoming engine or turbine while the other does the switching at the board. In some installations provision is made for controlling the governors electrically from the switchboard. There are, however, many smaller "one man" plants in which the problem of getting two machines in synchronism is less simple.

With the present prevalent method of switching it is necessary for the operator, when cutting a new machine onto the line, to adjust the speed of the incoming alternator to obtain exactly correct frequency, then to adjust the voltage and await opportunity to close his main switch when the frequency and phase correspond with busbar conditions as indicated by lamps or a synchroscope. A fluctuating load causes slight changes in line voltage or frequency. Also it is difficult to maintain a constant speed on the incoming machine running without load. For

these reasons the synchronizing operation is often a rather difficult and tedious one.

A switching arrangement has recently been tried out which permits cutting in an alternator without the necessity of first obtaining a synchronous condition. The principle involved is merely that of throwing a single phase of the incoming machine onto the busbars with a sufficient reactance inserted in series to choke down the initial rush of current. The so called synchronous impedance of the armature circuit is increased and the reduced cross-current quickly draws the machine into step. The main switch can then be closed and the reactance cut out of the circuit. The result is accomplished in a manner free from jerk or jar and without detrimental effect upon the busbar voltage.



SWITCHES AND CONNECTIONS FOR PUTTING TWO GENERATORS IN PARALLEL

The circuits may be arranged as shown. Double-throw oil switches with the interlocking devices removed may be used readily, although for simplicity in the illustration ordinary knife switches are shown. In switching in a new machine the generator is driven at approximately correct speed and the voltage adjusted to correspond with that of the busbars. The left-hand switch on the incoming generator panel is then closed, throwing the single phase on the line with the series reactance inserted. If synchronizing lamps are provided they will quickly indicate synchronism. The main right-hand switch may then be closed and the left-hand switch opened. The rheostat of the incoming machine is then adjusted to obtain a minimum wattless current.

It will be seen that with this method of operation it is almost as simple to parallel alternators as direct-current machines. The method has been tried successfully in several plants. It has been found possible for one man to start a unit from rest and place it in operation on the busbars in less than one minute. The simplicity and effectiveness of the system should recommend its general use for small plants.

CORRESPONDENCE

Generator Trouble

The trouble with Mr. Landmesser's generators mentioned in the Oct. 15 issue is probably due to the compounding, assuming the engine regulation to be good, and is caused by varying voltage due to a change of load. As the load changes the voltage may run up on one machine sufficiently to allow the current to flow into the other generator. The most likely remedy would be to adjust the shunts on the series winding so there will be less variation in the voltage as the load changes. It is hardly to be expected that the series winding on the large machine is connected in opposition to the shunt winding which would tend to reduce the voltage as the load increased.

There are automatic voltage regulators on the market, but leads for these would have to be run between generators to cause the regulators to operate.

It might be well to see that trolley lines and feeders do not make temporary connection with other wires carrying high-voltage currents. I had the same trouble, caused by a lighting line coming in contact with a trolley wire every time a car passed a certain spot. The trouble was not located until one Sunday when the plant was shut down and the voltmeter needle was seen to move. An extended investigation of the neighborhood soon showed the cause, and when this was removed the trouble ceased.

Of course, it may be that the engines do not regulate as quickly as they should, and thus allow the voltage to rise or fall as the load varies, or a poor field connection might cause the machine to be slow in building up when a heavy load came on.

THOMAS H. WATSON.

Chicago, Ill.

What Reversed the Motor?

Referring to the letter in the Oct. 29 issue under the title "What Reversed the Motor?" an examination of the diagrams will show that, probably due to an oversight, the resistance and lamp are shown in one of the main leads whereas the article states they were inserted in the field circuit. The fact that the motor ran satisfactorily with the resistance in circuit and would reverse and run at a higher speed when the 50-watt lamp was substituted would show that with the lamp in series the magnetizing effect of the field winding was reduced to such a low value that the field set up by the armature reaction demagnetized it. With the brushes set with a backward lead the armature reaction would set up a field for the armature currents to react on in an opposite direction to that set up by the field winding, causing the motor to reverse.

With the brushes set with a forward lead it would continue to run in the same direction but with a motor of any considerable size the sparking would be severe. Mr. Boone will find that with his field circuit open entirely and a forward or backward lead to his brushes, by passing current through his armature alone it will revolve, although, due to the weak field, the torque developed would be very small.

If circumstances warrant the expense I would suggest that the field winding be changed to a regular shunt. The data for this purpose could be obtained as follows: With the motor connected to the line and running at normal speed and full load, measure the current with an ammeter in series. In the case of a series motor this will measure the current through the field. Obtain the number of turns in the series field. The number of turns times the current will equal the ampere-turns required in the new shunt winding. Allowing about 1200 circ. mils per ampere; the size of the wire may be readily calculated.

P. JUSTUS.

Cleveland, Ohio.

Blown Fuse

In answer to Mr. Koppel's question as to which fuse was blown under the heading of "Blown Fuse" in the Nov. 5 issue, I would say that as the middle transformer which is fed from the line wires *B* and *C* delivered its proper voltage to the motor, it must have been the fuse in line *A* which was blown.

F. S. PIPER.

Manchester, N. H.

Gas Power Department

Worth-while gas-engine and producer information treated in a way that can be of practical use

Operating Gas Engines

By A. L. Brennan, Jr.

Although internal-combustion engines have reached a high stage of perfection as far as design and construction is concerned, still as regards their running qualities, much naturally depends upon the operator. Unless he is versed in at least the fundamental principles of these self-contained power-generating units the efficiency will be very low.

Trouble is often encountered when starting a gas engine by its failure to pick up. In such cases faulty gas mixture or ignition troubles are usually to blame. The former is sometimes caused by insufficient fuel supply, due to a valve being closed, the pipe clogged, or the vent closed. The latter may be due to an open circuit, to poor compression, to water in the cylinder due to a leak in the jacket or to moisture sucked in from the exhaust.

Low-tension ignition of the make-and-break type should give but little trouble, yet it is well to know how to trace a trouble in this circuit. To ascertain at once if the existing trouble is in the wiring or igniter, place the switch in position to close the circuit and then with a screw-driver form a circuit from the stationary electrode on the igniter to the cylinder. If a good spark results by moving the screw-driver, the circuit is complete and the trouble is in the igniter. Having traced the trouble to the igniter, it may be due to dirty points or to the actuating springs working improperly.

On the other hand, if no spark is had it will be necessary to look over the battery connections and terminal points of all the wiring, for an open circuit is in nine cases out of ten responsible, as the low-tension current is not as apt to short-circuit as in the case of the high-tension or jump-spark system.

Briefly considering ignition troubles pertaining to the high-tension system it will be found that when the engine refuses to start, stops suddenly or misses fire, it is best to trace the trouble by a process of elimination. This is accomplished by turning the engine over by hand and noting if each section of the coil buzzes. If each vibrator works all right, the trouble cannot be in the primary side of the system, that is, in the batteries, timer battery connections, primary winding of the coil or contact points on the vibrator; hence the only parts remaining are the high-tension wires and the spark plugs.

To test the high-tension wires to the plug, hold the end of the wire about $\frac{1}{16}$ in. from the plug with the engine on the firing point for that cylinder, and switch on the battery. If a good spark jumps across the gap the existing trouble has been traced to the spark plug. This may be tested by first removing the plug from the cylinder, wiring as before and placing the threaded part on the cylinder. Throw on the switch, and if no spark appears, substitute another plug for the same test to prove whether this is the true location of the trouble. Plug troubles are sometimes overcome by merely cleaning the points.

On the other hand, if upon turning the motor to the firing position no buzz is had, any one of the following causes may be looked for: A disconnected ground wire from the coil to the engine, a broken or disconnected wire in the battery box, poor contact at the timer, a broken connection inside of the coil, vibrator contact points out of adjustment or badly pitted, or the coil otherwise wet or damaged, weak batteries or a dead cell.

Again, assume that some, but not all of the sections buzz. In that case look to the contact points of the sections not buzzing, and to the wire from that section to the timer. If this does not show the trouble look for an internal defect in that section. If the units of the coil are interchangeable the last point may be positively determined by removing the faulty section and replacing it with another that is known to be in working order. Unless one is familiar with coils, however, or the conditions warrant an attempt to remedy a defective coil, the unit should be sent to the manufacturer or some other reliable place. In cases where all the sections buzz, but no spark appears at the plugs the trouble can usually be traced to a loose connection on the secondary side of the coil.

To clean a spark plug thoroughly it should be taken apart and washed with gasoline, the points being trued with a fine flat file and brightened with emery cloth. Defective insulation allowing partial short-circuits and other troubles are of a wide range, but if one becomes familiar with the general troubles the special ones will be easily found.

Coils should be adjusted when the motor is in operation, and the tremblers should be allowed only sufficient tension to produce positive ignition. Many operators have the idea that sparking efficiency can be greatly increased by considerable tension on the tremblers. However, this only to a certain extent holds true and it is found that a stiff adjustment of the contact points results in quickly running down the batteries, pitting the contact points and impairing the efficiency.

Engines operating in an irregular manner will not give high efficiency; hence it is better to shut down for a time, if possible, to remedy the trouble, if it cannot be corrected while the machine is in operation. The chief factors in the troubles of this kind are water in the carburetor, faulty mixture, gasoline feed partly choked, loose terminals, broken-down insulation on the wires, faulty contact at the coil or timer, dirty or defective spark plugs, or mechanical faults causing poor compression, such as badly pitted or broken valves.

If the cylinders fire in order, but the engine does not develop its full power, the cause may be faulty mixture, insufficient lubrication, poor compression, exhaust partly obstructed, or defects in the ignition system.

Overheating is usually due to some failure in the cooling system, although insufficient lubrication will produce the same result. If the engine is operating in a weak manner and apparently the cooling system is not at fault, the trouble may be due to a crack in the piston head or to burned gas getting past the piston because of worn rings or a scored cylinder. Either of these will cause the crank case to become hot, which will help in locating the trouble.

Hissing sounds are in nearly every instance an indication of the compressed gas escaping, which may be due to a broken spark plug, a partly blown-out gasket, open compression cocks, etc. A sucking sound is invariably the result of the intake manifold being loose. If this fault is not corrected the mixture will be weakened by the addition of air, and the engine will lose power, although it may appear to operate regularly.

Muffler explosions are frequently the result of the gas mixture being too lean to fire in the cylinder, although they may also be caused by an inefficient spark, over-retarded spark, or to pitted or stuck exhaust valves. When a motor slows down gradually accompanied by misfiring it is nearly always an indication that

the gasoline supply is at fault. The carburetor may be choked by dirt at the jet, the gasoline tank may be empty or air bound, or the gasoline valve may be partly closed. Weak batteries or dirty spark plugs will also contribute to this. Sudden stopping is nearly always due to the ignition giving out, such as a broken wire, loose terminal, broken spark plug, trembler on spark coil stuck, faulty contact at the timer, switch out of contact or no gasoline.

Explosions in the carburetor or inlet pipe are often directly attributed to a lean gas mixture or mechanical failures such as the inlet valve or spring or the valve itself getting out of time; or the spark too far retarded.

Overheated exhaust often indicates a clogged pipe or an over-retarded spark. Knocking or pounding can usually be traced to a loose flywheel, loose cylinder or crank case, loose or worn bearings or to the spark too far advanced, too rich a mixture or defective lubrication which will at first produce a squeaking before the surface is badly worn.

Undoubtedly to some, the foregoing remarks will give an impression that the gas engine is still subject to the many complications of former years. This to a certain extent holds true but, at the same time, it should be realized that the greater majority of the present-day troubles can be traced directly to carelessness or neglect on the part of the operator.

A large percentage of engine failures as far as operation is concerned can be directly attributed to over or poor lubrication of the cylinder. Too large a quantity of high-grade cylinder oil will produce the same results as the correct amount of poor oil; that is, it will gum up the piston rings, produce excessive carbon, and cause dirty points on the igniter or spark plugs. In fact, low flash point oils are little better than useless when subjected to the high temperatures of an internal-combustion motor.

These points are mentioned to lay stress upon the importance of looking carefully after the details and thus avoiding the complications which are bound to ensue. As soon as an operator realizes the importance of paying strict attention to methods others have found to be the best, his own troubles will become a thing of the past.

The Junkers Engine—V

By F. E. Junge

Pressure on Guides

To arrive at a complete understanding of the action of forces in Junkers and Diesel engines, respectively, it remains to study the pressure on the guides and the resulting pressure on the main bearings, as well as the inertia forces and tilting moments. The different pressures on the crosshead guides, resulting from those previously demonstrated are shown in Figs. 24 and 25. It is apparent that in the Junkers engine the pressure on the guides experiences a hardly perceptible reversal toward the end of the stroke. In the Diesel engine, however, the crosshead is lifted from its guides during the second half of the compression stroke, because of the change of direction of the pressure on the guides. For a definite rotation, therefore, wear takes place on both guides. This is much enhanced due to impact as soon as there is enough play between the crosshead and its guides. If the piston is to serve as a crosshead, there will be wear on both sides of the piston and the cylinder; both will experience about four times the wear on one side. This will lead to leakage and the loose fit of the piston will augment undue wear.

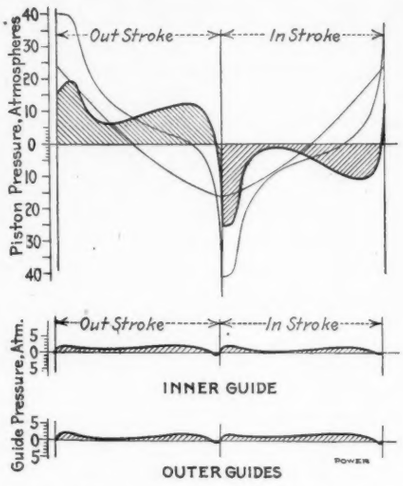


FIG. 24. RESULTANT PRESSURES IN JUNKERS ENGINE

Pressure on Main Bearings

In the Junkers engine the load on the main bearing is made up of the unbalanced inertia forces, acting along the center line of the engine and the pressure on the guides, at right angles to the center line. In the following it is assumed that the weights of both mechanisms are the same; these conditions conforming with practice.

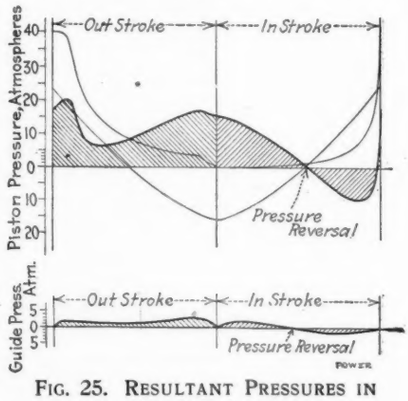


FIG. 25. RESULTANT PRESSURES IN DIESEL ENGINE

The magnitude and nature of the inertia forces are graphically shown in Fig. 26. That such forces should occur is due to the fact that the opposite motion of the pistons in the Junkers engine is not identical at all points of the stroke. The arrangement of two connecting-rods on the same side of the crank circle brings about the opposite motion of the pistons. But the inertia forces, due to the obliquity of the connecting-rod, augment one another and are responsible for the remaining unbalanced force. The inertia forces acting on both the middle and outside pistons have been plotted. It should be noted that when one set of pistons is on the forward or out stroke, the other is on the inward or back stroke. The two inertia curves are superimposed, being referred to the travel of the middle mechanism. The inertia forces neutralizing each other are shown cross-hatched in Fig. 26.

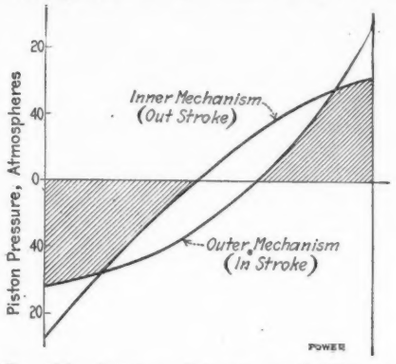


FIG. 26. INERTIA FORCES IN OUTER AND INNER MECHANISM

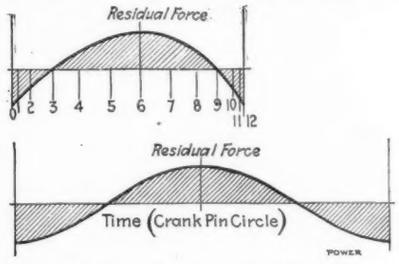


FIG. 27. UNBALANCED FORCES

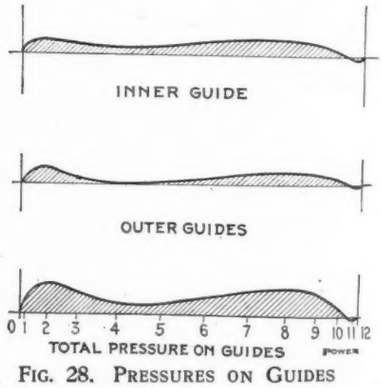


FIG. 28. PRESSURES ON GUIDES

Fig. 27 again represents the unbalanced forces; in the upper diagram they are referred to the travel of the center mechanism and in the lower to the crankpin circle. As may be seen from the diagrams, the remaining force completes its cycle during one stroke. This corresponds with its character as the sum of force errors which in first approximation travel at twice the rate peculiar to the fundamental acceleration term for an infinitely long connecting-rod. With a connecting-rod of a length equal to that of five cranks, the maximum value of the unbalanced force is two-fifths of the inertia force in the dead center position with an infinitely long connecting-rod. If the latter be 295 lb. per sq.in., the maximum remaining force will attain a value of 118 lb. per sq.in., in this case.

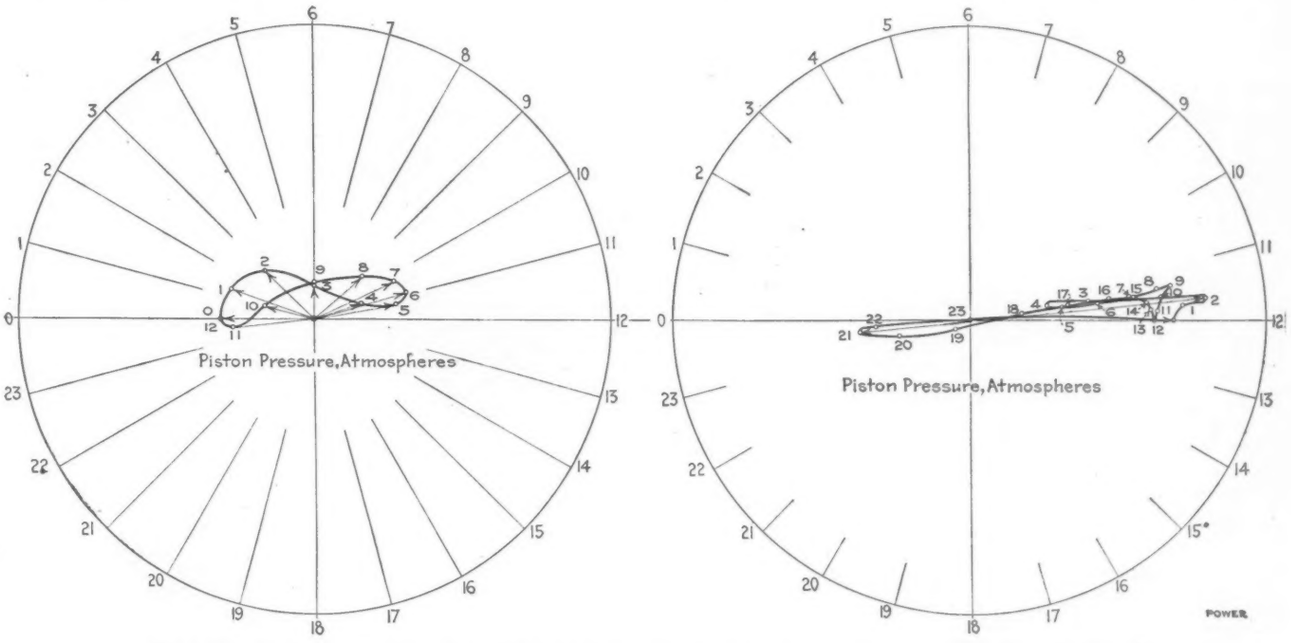
The nature of the pressure on the guides is graphically represented in Fig. 28, showing the pressures on the inner and outer guides, respectively, and the combined pressure of the two.

Finally the resulting load on the main bearing is represented in Fig. 29, for successive positions of the crank. This is simply the geometrical resultant of the corresponding unbalanced force in the direction of the cylinder axis, and of the pressure on the guides, acting at right angles to the center line of the engine, the total bearing presssure being given.

For the Diesel engine the resulting pressure on the main bearing is made up of the gas pressure on the piston, the inertia forces, coinciding in direction with the center line and the pressure on the guides, at right angles to the latter. The first named force results from a compensation of the working stresses not being effected within the crank mechanism; hence the main bearings are exposed to the full working stresses, as was discussed in Article IV.

Fig. 30 shows the corresponding load on the main bearing for corresponding positions of the crankpin. The resulting pressure on the bearing of the Diesel engine is more than that in the case of the Junkers engine. Moreover, the load on the bearing is fluctuating in the Diesel engine, whereas in the Junkers engine it is almost uniform. The changing of the loads supported by the bearings induces an unequal application of the pressure and unequal wear. The weak and rapid force undulations in the Junkers engine are totally eclipsed by the slow and strong force fluctuations in the Diesel engine. It should be further noted that when starting up, the pressures on the bearings are about twice those indicated, because the inertia forces are then not acting compensatingly.

Regarding the action of forces in an engine it is important to consider not only their effect upon the engine itself, but to reflect also upon their effects on the surroundings. This should be done quite aside from whatever the conditions set up in the engine itself by these forces may be, or how they are taken up by the separate elements, and the magnitude of the resulting stresses. In stationary engines it is generally possible to provide foundations substantial enough to cope effectively with prevailing conditions. In marine engines, however, this matter commands more detailed consideration, because of the greater complexity of the actions concerned. This is especially so when several separate units are to form one compact aggregate of machinery.



FIGS. 29 AND 30. RESULTANT BEARING PRESSURES IN JUNKERS AND DIESEL ENGINES RESPECTIVELY

CORRESPONDENCE

Producer Operation

Gas-engine users, also steam engineers, are commencing to have more respect for the gas engine than they formerly had, and realize that it requires as much skill to be a successful gas-engine operator as it does to run a steam engine. Furthermore, a producer operator requires as much knowledge as a fireman. Our greatest trouble was to get the producer operators to thoroughly poke the fuel bed and to break up clinkers. However, when they learned that thorough poking keeps the fire clear and the clinkers from forming, they took care that none formed. As long as the fuel bed is in proper condition it does not require any hard poking; also the proper adjustment of the air and steam is soon found giving a good quality of gas.

Our producers are given a thorough poking down with a heavy bar before starting, and after blowing 15 to 20 min. the fuel bed is tamped with a tamping bar, and then after blowing a few minutes we have a good quality of gas. We always keep a charge of coal in the hopper when running or idle (in our case 100 lb.), also water running through the stack valve when blowing, 15 min. after shutting down and, of course, while running. Our producers have not had the fire out in five years, they run an average of 10 hr. a day, a 100-lb. charge of pea coal being dumped on the fire according to the demand, and also upon shutting down. The ashes are removed every other day or as the fuel bed reaches a height, generally about 4 ft. thick.

Middleboro, Mass. J. W. FRIES.

Heating and Ventilation

Considered as power-plant problems. Layout and operation of systems and apparatus

Cost of Exhaust Steam Heating*

By Ira N. Evans†

"Our heating costs us nothing, when our power is operative, as we utilize exhaust steam entirely."

This statement is frequently made by owners and engineers, and it is thought by many to be universally true. That there is a saving by utilizing the latent heat of the exhaust steam after it has been expanded in an engine is unquestioned, but has the maximum power been obtained before the steam enters the heating system?

The heat represented by work done by the steam engine is only about 10 per cent. of the amount introduced into the fluid to make it available for power purposes, while 90 per cent. can be used for heating or goes to waste through the injection water of the condenser. All additional power requires 100 per cent. of the heat, while 10 per cent. only can be utilized, so that it is all the more imperative that this feature of the problem should receive close attention.

The heating system should not only require a minimum weight of steam, but be capable of utilizing the steam at the lowest temperature possible, in order that it can be expanded between wider limits of temperature and thus increase the engine efficiency. This statement holds true regardless of the type of engine, although some give better efficiency for wider ranges of temperature than others, or method of heating.

Heating and power are seldom properly correlated. In many cases the power plant is designed independently and the heating system left as an after consideration. The more uneconomical the power equipment the greater will be the apparent saving by utilizing the exhaust steam for heating, and the greater the cost of heating and power combined. The real saving is the amount of steam utilized for the power load under the most economical conditions of full vacuum. All steam additional due to the higher temperature of the exhaust is a charge against the heating system. This charge is still there even if the engines are modified and made more uneconomical to suit the heating system. A condensing engine may be operated noncondensing to suit the heating system and the power balance the heating, showing a large saving, but the cost of the heating is still the difference in the amount of steam required for operation at the final temperature conditions. Incidentally the rated capacity of the machine is reduced as the temperature range of the steam decreases.

It will be evident that hot-water heating only is applicable for operation under temperatures to any extent below atmosphere. The water system should be designed and operated, so that the maximum power will be obtained from the steam before it enters the heating system, which should be made as efficient a condenser as possible. This means the lowest temperatures of steam and water compatible with the initial cost of radiation, as the lower the temperature at which the steam is utilized the more costly the heating system. If the heating system requires high exhaust-steam temperatures or the engines are operated under conditions of lower vacuums or atmosphere, the economy of the combination is reduced.

The principle outlined applies to all exhaust-heating systems, and most forcibly to central stations and plants where there are periods of surplus peak load.

If the heating load is considerably in excess of the power load it makes little difference in the total economy whether or not economical engines are installed. The only question then involved is to select a heating system using the minimum weight of steam.

Excess heating loads are seldom found in large plants or central stations and even if they do exist, there is always the probability of the power load increasing. For the most economical results the heating load should not be greatly in excess of the power load.

In the issue of Sept. 3, 1912, the heating plant at Oak Park was described and several schedules of operation given. A schedule averaged from these and showing a maximum drop of 66 deg. in 20-deg. weather is used in this article to show the effect and advantages of doubling the pumpage and reducing the maximum drop to 32.5 deg., or to approximately one-half. The discussion does not apply to this particular plant, as it is a live-steam proposition and the mains are not arranged to work successfully with high velocities.

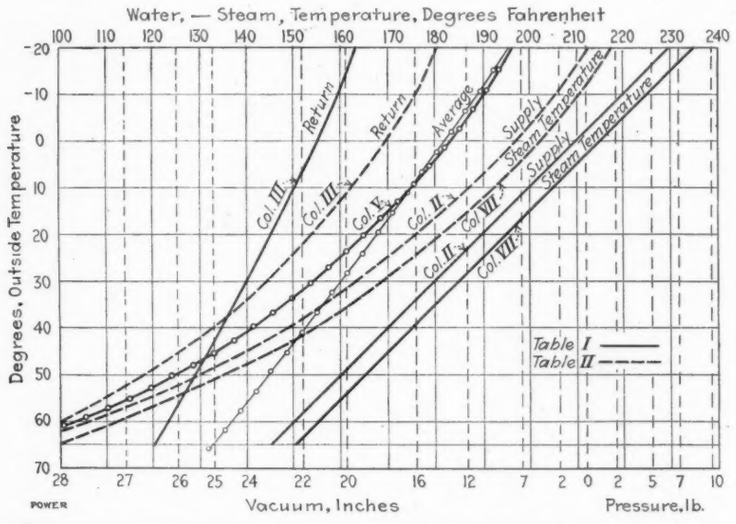


Fig. 1. Temperatures of Steam and Water for Range of Outdoor Temperatures

Tables 1 and 2 and Fig. 1 show the temperature conditions of operation with the pumpage as at present and with it doubled. Supply and return temperatures, average temperatures of water and required steam temperatures are all given in separate columns, and the same Roman numeral at the head of the column refers to the same quantity in either table and in the curves of Fig. 1. The steam temperature was assumed to be in all cases 5 deg. higher than the outboard temperature of the water. For the Oak Park plant the amount of heating surface is

*Copyrighted, 1912, by Ira N. Evans.

†Consulting engineer, 156 Broadway, New York City.

TABLE 1. OPERATING CONDITIONS BEFORE DOUBLING THE PUMPAGE

I. Outside Temperature Deg. F.	II. Supply Temp. Deg. F.	III. Return Temp. Deg. F.	IV. Drop, Deg. F.	V. Average Water Temp. Deg. F.	VI. Average Water Temp. — 70 Deg. F.	VII. Temp. Steam in Heater	VIII. Cor. Press. or vac.	IX. Steam Rate per kw.-hr. (500 kw. unit)	X. Pounds Water Circulated Per Min.	XI. Steam per Hour Used on System (Latent heat 1000 B.t.u. per lb.)	XVI. Boiler hp. at 34.5 lb.	XII. Head on System in Feet	XIII. Power on Pump, kw. (70% Eff.)	XV. Net Available kw. (pump deducted)
—20	229	163	66	196	126	234	8 lb.	49	30,810	122,008	3537	112	112	2377
—15	224	161.5	62.5	193	123	229	6	47	30,500	114,375	3317	109.3	109	2324
—10	219.5	160	59.5	190	120	225	5	46	30,500	108,885	3158	109.3	109	2258
—5	215	158	57	186.5	116.5	220	3	43.5	30,500	104,310	3024	109.3	109	2291.
0	210	156	54	183	113	215	1	42	30,500	98,820	2866	109.3	109	2244
5	205	153	52	179	109	210	0	41.5	30,500	95,160	2760	109.3	109	2184
10	200	150	50	175	105	205	2 in.	40.5	29,900	89,700	2690	105	100	2150
15	195	147	48	171	101	200	6	38	29,900	86,112	2497	105	100	2166
20	190	145	45	168	98	195	8	36	29,900	80,730	2341	105	100	2143
25	185	141	44	163	93	190	10	35	28,100	74,184	2151	93	85	2034
30	180	130	42	159	89	185	12	34	28,100	70,812	2054	93	85	1998
35	175	136	39	156	86	180	14	32.5	28,100	65,754	1907	93	85	1938
40	170	134	36	152	82	175	16	31	28,100	60,696	1760	93	85	1873
45	165	131.5	33.5	148	78	170	17	30.2	26,400	53,064	1539	81.4	70	1687
50	160	129	31	145	75	165	19	29	26,400	49,100	1424	81.4	70	1623
55	155	126	29	141	71	160	20	28	26,400	45,936	1332	81.4	70	1571
60	150	123	27	136.5	66.5	155	21	27	26,400	42,768	1240	81.4	70	1514
65	145	120	25	132.5	62.5	150	22	26.5	26,400	39,600	1148	81.4	70	1424
						100	28	20						

I. Outside Temperature Deg. F.	XIV. B.t.u. per hr. per sq.ft. per deg. dif. (570,000 sq.ft.)	XVIII. Returned in Friction, lb. Steam	XIX. Cost Pumpage 160 lb. to vac. Col. VIII Steam Rate Col. IX	XVII. Dif. in Total Heat 160 lb. and Col. VII	XX. Steam to Raise Feed to 212 Deg. F.	XXI. Corrected Steam Rate Col. IX + Col. XX	XXII. Cost of Heating per kw.-hr. Col. XXI—22.24	XXIII. Cost of Heating Over full Vacuum Col. XV × Col. XXII	XXIV. % Time Each Temperature Period	XXV. Hours Each Temp. Period
—20	1.671	266	203	37		49	26.76	63,609	0.05	0.5
—15	1.618	257	200	39		47	24.76	57,542	0.08	0.8
—10	1.6	257	200	40		46	23.76	53,650	0.14	1.4
—5	1.571	257	199	42		43.5	21.26	48,707	0.21	2.1
0	1.534	257	201	44		42	19.76	44,341	0.52	5.2
5	1.531	257	208	46		41.5	19.26	42,064	1.40	14
10	1.5	242	195	48	0.284	40.78	18.54	39,861	2.20	22
15	1.496	242	190	50	0.456	38.456	16.216	35,124	4.20	42
20	1.445	242	187	52	0.612	36.612	14.372	30,800	5.50	55
25	1.4	202	160.6	54	0.77	35.77	13.53	27,520	8.60	86
30	1.4	202	162	56	0.92	34.92	12.68	[illegible]	17.40	174
35	1.341	202	160	58	1.04	33.54	11.30	21,900	14.40	144
40	1.3	202	158	60	1.15	32.15	9.91	18,561	12.30	123
45	1.215	166	131	62	1.27	31.47	9.23	15,570	10.00	100
50	1.15	166	134	64	1.36	30.36	8.12	13,179	8	80
55	1.135	166	129	66	1.46	29.46	7.22	11,343	6	60
60	1.128	166	129	68	1.54	28.54	6.30	9,538	5	50
65	1.110	166	130	70	1.64	28.14	5.90	8,402	4	40
					2.24	22.24				

I. Outside Temperature Deg. F.	XXVI. Total Cost per Season lb. Steam XXIII × XXV	XXVII. Net Available kw.-hr. per season XV × XXV	XXVIII. Total lb. Steam for Heating and Power Col. XI × XXV	XXIX. Steam per hr. at 28 in. vac. Col. XV × 22.24 lb.	XXX. Total kw-hr. Pump.
—20	31,804	1,189	60,984	52,864	56
—15	46,034	1,859	91,500	51,686	234 hr.
—10	75,110	3,161	152,439	50,129	109 kw.
—5	102,285	4,811	219,051	50,952	2,562
0	230,573	11,669	513,864	49,907	119 hr.
5	588,896	30,576	1,332,240	48,572	100 kw.
10	876,942	47,300	1,973,400	47,816	11,900
15	1,475,208	90,972	3,616,704	48,172	527 hr.
20	1,694,000	117,865	4,440,150	47,660	85 kw.
25	2,366,720	174,924	6,379,824	45,236	44,795
30	4,408,290	347,652	12,321,288	44,436	330 hr.
35	3,153,600	279,072	9,468,576	43,101	70 kw.
40	2,283,003	230,379	7,465,608	41,655	23,100
45	1,557,000	168,700	5,306,400	37,519	
50	1,054,320	129,840	3,928,000	36,096	
55	680,580	94,260	2,756,160	34,939	
60	476,900	75,700	2,138,400	33,671	
65	336,080	56,960	1,584,000	31,670	
Totals	21,437,345	1,866,889	63,747,588		82,413

TABLE 2. OPERATING CONDITIONS AFTER DOUBLING THE PUMPAGE

I. Outside Temp. Deg. F.	II. Supply Temp. Deg. F.	III. Return Temp. Deg. F.	IV. Drop Deg. F.	V. Average Water Temp., deg. F.	VI. Average Water Temp. — 70 deg. F.	VII. Temp. Steam in Heater	VIII. Cor's'p'd Press. or Vacuum	IX. Steam Rate Per kw-hr. (500 kw. unit)	X. Pounds Water Circulated Per Minute	XI. Pounds Steam per Hour on System	XII. Head on System, ft.	XIII. Power on Pump, kw. (70% Eff.)	XIV. B.t.u. Per Hour per sq.ft. per Deg. Dif. (510,000 sq.ft.)	XV. Net Available kw. (Pump Deducted)
—20	212.5	180	32.5	196	126	21[illegible]	1.5 lb.	43	61,200	119,340	100	200	1.86	2575
—15	209	178	31	193.5	123.5	214	0.5	42	61,200	113,832	100	200	1.81	2510
—10	206	176.5	29.5	191	121	211	0	41.5	61,200	108,324	100	200	1.76	2410
—5	202	173.5	28.5	188	118	207	3 in.	40	61,200	104,652	100	200	1.74	2416
0	196.5	169.5	27	183	113	202	5	38	61,200	99,144	100	200	1.72	2409
5	192	166	26	179	109	197	8	36	61,200	95,472	100	200	1.717	2452
10	187.5	162.5	25	175	105	193	10	35	61,200	91,800	100	200	1.714	2423
15	181.7	158.2	23.5	170	100	187	12	34	61,200	86,292	100	200	1.69	2338
20	176	154	22	165	95	181	14	32.5	61,200	80,784	100	200	1.67	2285
25	168	148	20	158	88	173	17	30.5	61,200	73,440	100	200	1.634	2208
30	163	144.5	18.5	154	84	168	18	30	61,200	67,932	100	200	1.624	2064
35	157	139.5	17.5	148	78	162	20	28	61,200	64,260	100	200	1.62	2095
40	150	134	16	142	72	155	21.5	27	61,200	58,752	100	200	1.606	1976
45	140	126	14	133	63	145	23	25.8	61,200	51,408	100	200	1.606	1792
50	130	118	12	124	54	135	25	25	61,200	44,064	100	200	1.60	1562
55	117	107.5	9.5	112	42	122	26	22.5	61,200	34,884	100	200	1.591	1350
60	106	99	7	102	32	111	27	21	61,200	25,784	100	200	1.58	1024
65	93	88.5	4.5	91	21	100	28	20	61,200	16,524	100	200	1.54	626

I. Outside Temp. Deg. F.	XVI. Boiler hp. at 34.5 lb.	XVII. Dif. in Total Heat 160 lb. and Col. VII	XIX. Cost Pumpage 160 lb. to vac. Col. VIII Steam Rate Col. IX	XVIII. Lb. Steam Returned in in Friction	XX. Steam to Raise Feed to 212 Deg.	XXI. Corrected Steam Rate Col. IX + Col. XX	XXII. Cost of Heating per kw. Over full Vac. Col. XXI—22.24	XXIII. Cost of Heating Over full Vac. Col. XV × Col. XXII	XXIV. % Time Each Temp. Period	XXV. Hours Each Temp. Period
—20	3460	44	379	472		43	20.76	53,457	0.05	0.5
—15	3300	45	378	472		42	19.76	49,598	0.08	0.8
—10	3140	46	382	472		41.5	19.26	46,417	0.14	1.4
—5	3033	47	376	472	0.2	40.2	17.96	43,391	0.21	2.1
0	2874	49	372	472	0.38	38.38	16.14	38,881	0.52	5.2
5	2767	51	367	472	0.54	36.54	14.3	35,064	1.4	14
10	2661	53	371	472	0.665	35.665	13.425	32,529	2.2	22
15	2501	55	374	472	0.85	34.85	12.61	29,482	4.2	42
20	2342	58	377	472	1.00	33.5	11.26	25,779	5.5	55
25	2129	61	372	472	1.116	31.616	9.38	20,711	8.6	86
30	1969	63	378	472	1.29	31.29	9.05	18,679	17.4	174
35	1863	66	370	472	1.37	29.37	7.13	14,937	14.4	144
40	1703	69	373	472	1.54	28.54	6.3	12,449	12.3	123
45	1490	73	377	472	1.73	27.53	5.29	9,480	10	100
50	1277	77	375	472	1.93	26.93	4.69	7,326	8	80
55	1011	83	374	472	2.025	24.525	2.285	3,085	6	60
60	745	88	370	472	2.12	23.12	0.88	904	5	50
65	479	92	368	472	2.24	22.24	0		4	40

I. Outside Temp. Deg. F.	XXVI. Total Cost Season lb. Steam XXIII × XXV	XXVII. Net Available kw.-hr. Per Season XV × XXV	XXVIII. Total lb. Steam for Power and Heating XI × XXV	XXIX. Steam per hr. at 28-in. Vac. Col. XV × 22.24 lb.	XXX. Kw.-hr. for Pump
—20	26,928	1,288	59,670	57,268	200,000
—15	39,678	2,008	91,065	55,822	200,000
—10	64,984	3,374	151,654	53,598	200,000
—5	91,121	5,074	219,769	53,732	200,000
0	202,191	12,527	515,549	53,576	200,000
5	490,896	34,328	1,336,608	54,532	200,000
10	715,638	53,306	2,019,600	53,888	200,000
15	1,238,244	98,196	3,624,264	51,997	200,000
20	1,415,095	125,675	4,443,120	50,818	200,000
25	1,781,146	189,888	6,315,840	49,106	200,000
30	3,250,181	359,316	11,820,168	45,903	200,000
35	2,150,928	301,680	9,253,440	46,593	200,000
40	1,531,227	243,048	7,226,496	43,946	200,000
45	948,000	179,200	5,140,800	39,854	200,000
50	586,040	124,960	3,525,120	34,739	200,000
55	185,100	81,000	2,093,040	30,024	200,000
60	45,050	51,200	1,285,200	22,774	200,000
65		25,040	660,960	13,922	200,000
Totals	14,762,277	1,890,928	59,782,363		

given as 550,000 sq.ft. and the mains aggregate 85,000 sq.ft. of actual surface. For calculation it is assumed that the main and heating surface is equivalent to a load of 570,000 sq.ft.

An analysis of the present schedule shows that a portion of the heating surface must be shut off in moderate weather. This is indicated by the low transmission in Col. XIV, Table 1.

In Table 2 all surface will be considered in operation, the pumpage will be the same throughout and every arrangement made to operate on the lowest water temperatures. This will make the heating system so responsive to outside weather changes that automatic heat control and local regulation will be unnecessary. The drop in head was given for the Oak Park plant. From this and the average distance the writer estimated the pumpage. The discharges taken from the different heads are only relative, as undoubtedly those used for the schedule in Table 1 are higher than actually obtained on this plant, but they are near enough for the comparative purposes of the article. The intervals of temperature changes for different outside weather conditions were estimated from an actual installation where the pumpage was nearly the same per square foot as in Table 2.

The writer found a mathematical relation in the ratio of the drop to the average water temperature minus 70 deg. The pounds of water pumped per hour divided by the square feet of surface, which was 7.2 lb. in this case, multiplied by the above ratio must give the rate of transmission of the heating surface per degree per hour. Having the limits given, the intervals can be determined by trial and error. The drop depends on the pumpage and the writer had the actual rates of transmission per square foot per hour, determined from the records of the plant referred to. The actual amount of water pumped, as given in Table 1, was estimated at 30,800 lb. per min. and in Table 2 the pumpage was made 61,200 lb. per min., which will give one-half the drop and deliver the same number of heat units at the maximum average temperature. With this drop and increase in transmission due to the additional pumpage, 10 per cent. less heating surface is required, or in round numbers 510,000 instead of 570,000 sq.ft., which means a reduction of a like amount in the cost of installations to customers.

A maximum average water temperature of 196 deg. was used in both cases. It might be policy in some instances to reduce the radiation and run slightly higher water temperatures. The constant amount of radiation and the constant pumpage have to be so related to the drop that (the square feet of radiation) × (the average water temperature — 70 deg.) × (the transmission per square foot per hour) will exactly equal (the drop) × (the pumpage per hour) for each period of outside temperature. With these quantities determined the schedule can be made up as the drop and (average water temperature — 70 deg.) will give the supply and return temperatures of water and the corresponding steam temperature.

The head of 112 ft. given as the maximum differential on the Oak Park plant schedule is assumed to give with three 10-in. pipe circuits a discharge of 30,810 lb. per min., with about 100 kw. required for the pumpage. For 61,200 lb. per min. three 12-in. mains require 175 ft. head and 350 kw. for the same length of circuit. As the mains require too large a proportion of the power generated to circulate the water, the comparison has been based on three 14-in. O. D. pipes which will discharge 61,200 lb. per min. with 100 ft. head and 200 kw. See Cols. XII and XIII of Tables 1 and 2. Fig. 1 shows that the increased pumpage causes a reduction in the steam temperatures at which it may be used on the heating system of over 17 deg. for the entire heating season. See Col. VII of both tables. This comes directly from the reduced drop in temperature necessary on the system.

In Col. IX the steam rate per kilowatt-hour is given for the various vacuums on a 500-kw. turbine. To find the pounds of steam per hour, Col. XI, multiply the pumpage per hour by the drop, Col. IV, and divide by the latent heat, 1000 B.t.u. By dividing by 34.5 the boiler horsepower is found, Col. XVI. The transmission per square foot is given in Col. XIV.

The latent heat of the steam increases with the reduction in pressure. If the steam can be utilized at lower temperatures and pressures on the heating system the same number of heat units can be transmitted with a greater economy in increased kilowatt output. Columns IX give the maximum steam consumption for Tables 1 and 2 as 49 and 43 lb. per kw.-hr. with 8- and 1.5-lb. pressure above atmosphere, respectively. The maximum vacuum and minimum steam consumption in Table 1 are 23 in. and 26.5 lb. per kw.-hr. and in Table 2, 28 in. and 20 lb. per kw.-hr.

In the case of high vacuum the steam rates should be based on 212 deg. and the amount of auxiliary steam necessary to raise the steam temperature from that corresponding to the vacuum to 212 deg. should be determined. The temperatures in Col. VII are deducted from 212 deg. and the result multiplied by the steam rate in Col. IX and divided by 1000 B.t.u. This gives the correction in Col. XX, which is added to the steam rate in Col. IX, giving Col. XXI. In Col. XXII is given the actual cost of heating per kilowatt-hour. It is obtained by subtracting in each case 22.24 lb.; the rate per kilowatt-hour for 28-in. vacuum corrected for 212 deg., from the rates in Col. XXI. This shows that exhaust steam for heating is obtained at considerable expense and not for nothing as is generally believed. What is actually saved is the steam at the maximum vacuum for the load carried, and the more economical the engine the less the saving will be. The cost at — 20 deg. temperature outside for Table 1, Col. XXII, is 26.76 lb. against 20.76 for Table 2, while at 65 deg. outside temperature the cost is 5.9 lb. per kw.-hr. against nothing in Table 2.

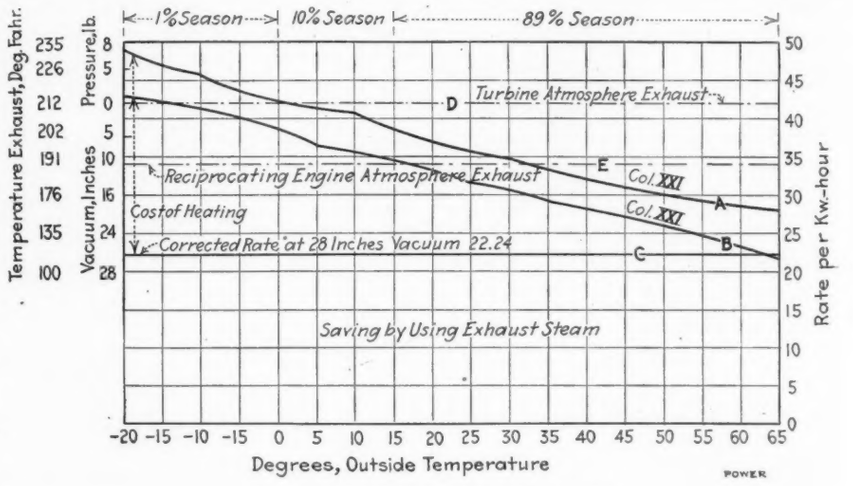


FIG. 2. RATE CURVES FOR THE SEASON

These costs are for hot-water heating under two conditions of operation, while if steam were used at 1½-lb. pressure the cost would be 20.76 lb. for each kilowatt load, whether the steam was needed or not, for the entire heating season.

The ordinates of curves *A* and *B* in Fig. 2, plotted from Col. XXI of both tables, give the corrected steam consumptions per kilowatt-hour. The percentages of time, Col. XXIV, for the different temperature periods are given and also the steam temperatures and correspond-

ing vacuums from Cols. VII and VIII. The ordinate under the line *C* gives the rate for a 28-in. vacuum as 22.24 lb. per kw.-hr., which is the actual amount of steam saved. All distances of ordinates above the line *C* are the charges per kilowatt-hour for heating due to reduced vacuum and are represented in Col. XXII of both tables. If the machine were operated at atmosphere the lines *A* and *B* would become *D* for the entire season. These steam consumptions multiplied by the load on the turbine will give the cost of heating and the total ordinate the cost of heat and power per kilowatt-hour whether the steam is all utilized or not. It may be readily seen that with a given heating load in pounds of steam that more power can be obtained from the steam rates of curve *B* than from the rates of curve *A*. The difference in rate curves is about 7 lb. per kw.-hr. for the entire season. The approximate line for the steam consumption of the reciprocating engine exhausting to atmosphere is indicated by line *E*, and line *D* gives the steam rate of a noncondensing turbine. A like chart may be made for any engine where the water rates are known for various vacuums. The temperature at which the steam is used in the heating system will determine the vacuum and the cost of utilizing the exhaust steam. The rate at full vacuum will determine the saving. The less the steam consumption at full vacuum, the greater the necessity of lowering curves *A* or *B* by operating the heating system at the lower steam temperatures. Fig. 2 will explain how owners think they are making large savings by utilizing exhaust steam on the heating and operating the engine on line *E*. They are actually saving the amount below line *C* if they are able to operate condensing at all.

With a 1000-kw., instead of a 500-kw., unit the saving due to increased pumpage would be appreciably increased on account of the greater economy of the larger machine at full vacuum. The cost of steam for the heating over the power load would also be increased.

Cols. XXIV and XXV give the percentage of time and number of hours probable for each period of operation out of a season of 1000. This is about the period of time during which the peak power load would occur with the heating in the central plant. The percentage of time each outside temperature period occurs has a marked bearing on the average rate for heating and power; that is, the average ordinate for Fig. 2 may not necessarily be the average for the season.

Assuming 70 per cent. efficiency for the pump in both cases and the same steam consumption as the main machine, the kilowatt load for the pumpage is given in Col. XIII. Against 112-ft. head the maximum power for a pumpage of 30,800 lb. is 112 kw. and 200 kw. for 61,200 lb. against 100-ft. head in three 14-in. O. D. pipes.

The actual heat units returned by friction to the water, reduced to pounds of steam, are given in Col. XVIII, Table 1, and calculated for Table 2, where the pumpage is constant, as follows:

$$\frac{61{,}200\,lb. \times 100\,ft. \times 60\,min.}{778\,ft.\text{-}lb. \times 1000\,B.t.u.} = 472\,lb.\,per\,hr.$$

The value 1000 B.t.u. has been used throughout for the latent heat per pound of steam, and, although somewhat in error, is near enough for the comparison when used in both cases.

Col. XVII of Tables 1 and 2 give the difference in total heat of the steam at 160-lb. gage and at the pressures of Col. VIII. Col. XIX gives the heat, reduced to pounds of steam, actually used in power for the circulation of the water. If the exhaust is utilized in the heating system, there will be no further loss from the pumpage. This is obtained as follows:

$$\frac{Col.\,XIII \times Col.\,IX \times Col.\,XVII}{1000\,B.t.u.} = Col.\,XIX$$

Col. XIX is less than Col. XVIII in both tables. This shows that where the available exhaust steam is less than the requirements of the heating, the pumpage, with 70 per cent. efficiency for the pump and less than 45 lb. of steam per kw.-hr., costs nothing. In the case of —20 deg. outside temperature it means for Table 1 that 5500 lb. of exhaust steam would come from the pumpage and in Table 2, 8600 lb. This is compulsory live-steam operation to that extent. The power load on the plant might be so balanced at times that it would be economical to reduce this live-steam operation by reducing the pumpage, which can be easily done with the same apparatus. At the same time reducing the pumpage increases the steam temperature of the exhaust and would increase the kilowatt rate appreciably on a large turbine fully loaded and naturally reduce its capacity.

The reduction in pumpage in Table 1 makes some of the periods slightly lower than for Table 2. This is probably caused by arbitrarily fixing the supply temperature when the operating schedule was obtained and trying to vary the heat supply with the pumpage. At the same time this plant as arranged and operated cannot be responsive to rapid changes in the radiation temperatures on account of the great difference between the supply and return temperatures.

It has been the writer's experience that there is no economy in reducing the pumpage in moderate weather, as the lower the average temperature of the surface and the smaller the drop the greater the vacuum can be carried, with a corresponding increase in power. The total heat required is also reduced. See in Fig. 3 the periods below 40 deg. outside temperature. It is possible where the vacuum is fairly high to carry considerable excess load over what would normally be required for heating on a turbine without appreciably interfering with the engine economy. This would not be possible if high-steam temperature were required for heating as the excess steam due to reduced vacuum might destroy all economy in the use of the exhaust steam.

Col. XI is divided by the steam rate in Col. IX and the kilowatts used on the pump deducted in each case to obtain the net kilowatts available for power, Col. XV. Fig. 3 is plotted with the quantities from Col. XXVI. Curves *A* and *C* are really a combination of three curves. Due to changes in the pumpage in Table 1 they become flatter and flatter as the pumpage is reduced, showing a decrease in the transmission. See Col. XIV. The sum of the squares between the full curves and the dotted curves is the cost of heating in each case. The squares

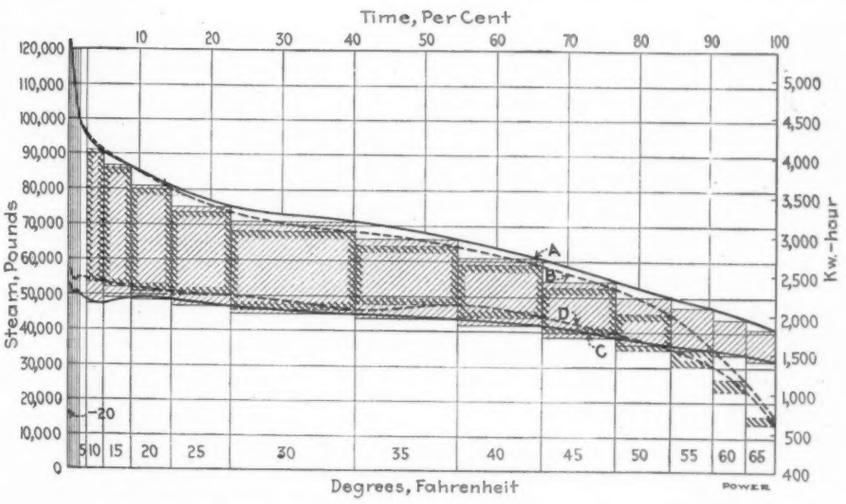


FIG. 3. POUNDS OF STEAM AND KILOWATT-HOURS FOR THE DIFFERENT TEMPERATURE PERIODS

below the curves *B* and *D* show the amount of steam actually saved, Col. XXIX of each table, and equals the sum of the kilowatt-hours in Col. XXVI multiplied by the rate at full vacuum. For Tables 1 and 2 the kilowatt-hours total up to 1,866,889 and 1,890,928 and multiplying by 22.24 gives 41,519,611 and 42,054,239 lb. of steam respectively. Thus 24,039 kw.-hr. were obtained from the steam used in Table 2 over Table 1, exclusive of the extra cost of pumpage.

The total areas for heating and power below the curves *A* and *B*, Fig. 3, amount to the totals of Col. XXVIII of both tables, or 59,782,363 lb. for Table 2 and 63,747,588 lb. for Table 1, a difference in favor of the latter of 3,965,225 lb. This is available for extra power at the rate of 22.24 lb. per kw.-hr. or 180,091 kw.-hr. The total increase in power will then be

$$180{,}091 + 24{,}039 = 204{,}130 \textit{ kw.-hr.}$$

or 10.8 per cent. increase in power by operation according to Table 2. The saving in actual steam, 3,965,225 lb., amounts to 6.22 per cent. The rate per kilowatt-hour for the power and heating, Col. XXVIII ÷ Col. XXVII is 34.146 lb. for Table 1 and 31.6 for Table 2, a difference of 2.546 lb. per kw.-hr. for the heating season of 1000 hr. This is greater than would be obtained by operating a condensing plant at 28 instead of 26 in. of vacuum.

The squares between the curves *A* and *C* of Fig. 3 for Table 1 and *D* and *B* for Table 2 show the cost of heating, or

$$63{,}747{,}588 - 41{,}519{,}611 = 22{,}227{,}977 \textit{ lb.}$$
$$59{,}782{,}363 - 42{,}054{,}239 = 17{,}728{,}124 \textit{ lb.}$$

Per kilowatt-hour this amounts to 11.91 lb. for Table 1 and 9.375 lb. for Table 2. This actually means the equivalent of live-steam operation in order to obtain the saving by the use of the exhaust steam. It is also the average ordinate for the cost of heating in Fig. 2 for the period of 1000 hr. This portion of live-steam operation in the case of Table 1 is 34.9 per cent. of the total steam used and 29.67 per cent. for Table 2. The proportion of live steam used in either case to the actual amount saved is

$$\frac{22{,}227{,}977}{41{,}519{,}611} = 53.5 \textit{ per cent.}$$

$$\frac{17{,}728{,}124}{42{,}054{,}239} = 42.15 \textit{ per cent.}$$

This shows a much larger proportion of live steam used in Table 1. The actual cost of heating on the net power obtained on the plant is shown in Col. XXVI and the totals divided by the net kilowatts gives the rate without the cost of pumping, or for Table 1, 11.483 lb. per kw.-hr. and for Table 2, 7.88 lb. per kw.-hr. These costs subtracted from the rates for total live-steam operation give the cost of pumpage per kilowatt-hour for Tables 1 and 2 as

$$11.91 - 11.483 = 0.427 \textit{ lb.}$$
$$9.375 - 7.88 = 1.495 \textit{ lb.}$$

As these rates are not much more proportionally than required to operate the pumps on a regular condensing outfit, and less than the quantities in Col. XX over 50 per cent. of the season, the author believes that the pumpage on a proper heating system and that for the condenser plant will be about the same and the true cost of heating would be, Col. XXVI, inclusive of the cost of pumping. If taken in this way there would be an added increase in power of the difference in the pumpage or

$$200{,}000 - 82{,}500 = 117{,}500 \textit{ kw.-hr.}$$

or nearly 10 per cent. additional.

The problem resolves itself into an adaptation of the heating system to perform the function of a condenser for the purpose of utilizing the latent heat with a minimum interference with the engine economy. The principles in condenser practice apply to the problem in hand.

In Fig. 3 the actual steam saved is not very much different in amount, due to the reduction in total requirements in weather below 30 deg. outside, but the net available power in Col. XV is about 10 per cent. more for the individual periods, notwithstanding the 100 kw. deducted for the extra pumpage.

Bleeder turbines and low-pressure turbines with reciprocating engines utilizing the steam at 5 lb. pressure are frequently used. This does not give any more economy or as much as an independent engine with 5 lb. back pressure for that portion of the steam used in the heating system. See Fig. 2 for the steam rate for 5 lb. pressure and the difference in the economy of the methods presented will be apparent. The economy of the heating steam for power purposes depends solely on the temperature and pressure to which it is expanded. The fact that there is a condenser beyond the bleeder turbine or the low-pressure turbine does not affect the power obtained from the heating steam in any sense. If the heating steam can be expanded beyond the 5 lb. pressure to 20 in. of vacuum, more power and more latent heat per pound will be obtained and in the case of hot water the heating system will act as a condenser, relieving the regular condenser of that much work.

This discussion relates solely to the use of turbine units for this purpose, although the same is true of the reciprocating engine in a lesser degree, and as the size of unit increases the question of decreased steam consumption on the heating becomes more important. See line *E*, Fig. 2.

If the engine consumption is increased 100 per cent., there is only 50 per cent. saving by utilizing the exhaust steam for heating. This makes it imperative with the increased economies in engine operation that the system of heating be adapted to the new conditions so as to interfere with the efficiency for power as little as possible. When engines used 40 to 60 lb. of steam per horsepower-hour, any steam system would do for the purpose without appreciable increase. With large units operating at 15 to 18 lb. per kw.-hr. on high vacuum the old methods will not do. The losses in mains are greater and the efficiency of the heating system as a condenser is less when using low pumpage.

The importance of considering the amount of power obtained from the heating steam as well as the temperature and weight of steam necessary for the heating system cannot be overestimated. If a heating system required 1000 kw. and it could be operated with 800 kw., or if 800 kw. were used at a high steam rate and 1000 kw. could be obtained from the steam, due to higher vacuum and lower temperatures of exhaust, there would be a saving of 200 kw. In a heating system this saving would be made seven months in the year.

It is important that the load balance the heating as with too heavy power load more steam will be used due to decreased vacuum than would be required if the heating were operated on live steam and the engines run under full vacuum. On the other hand, with the schedule of Table 2 the heating system can be operated on a turbine a large portion of the time with considerable excess load in moderate weather without serious reduction in economy due to the higher vacuum and lower steam temperature required. With a properly balanced load the curves in Fig. 2 show a reduction of 2 to 4 lb. net per kw.-hr. for the entire season, with 10 per cent. less radiation.

This discussion shows a saving in methods of operation on two hot-water plants without any actual increase in first cost of one over the other. If anything the arrangement indicated by Table 2 will cost less to install than that indicated by Table 1.

In time let us hope that the opening paragraph of the article may be modified to read: We are operating our engines under vacuum at all times, our heating system acts as a condenser in winter for what exhaust steam is required for heating and the cost of steam, due to reduced vacuum, is not over 40 per cent. of full vacuum conditions.

Work on the erection of the new $75,000 power plant of the Pittsburgh & Lake Erie R.R., at Dickerson Run, Penn., has been begun. The excavation work is well under way and the plant will be hurried to completion, as the railroad is in urgent need of electrical power to run the machinery of the hydraulic system and the new coaling and ash dumping. Cars will be loaded and engines coaled by machines and the ashes mechanically hoisted and dumped.

Issued Weekly by the
Hill Publishing Company
JOHN A. HILL, Pres. ROB'T MCKEAN, Sec'y.
505 Pearl Street, New York.
Monadnock Bldg., Chicago.
6 Bouverie Street, London, E. C.
Unter den Linden 31—Berlin, N. W. 7.
FRED R. LOW, Editor.

Correspondence suitable for the columns of POWER solicited and paid for. Name and address of correspondents must be given—not necessarily for publication.

Subscription price $2 per year, in advance, to any post office in the United States or the possessions of the United States and Mexico. $3 to Canada. $5 to any other foreign country.

Pay no money to solicitors or agents unless they can show letters of authorization from this office.

Subscribers in Great Britain, Europe and the British Colonies in the Eastern Hemisphere may send their subscriptions to the London Office. Price 21 Shillings.

Entered as second class matter, December 20, 1910, at the post office at New York, New York, under the Act of March 3, 1879.

Cable address, "POWPUB," N. Y.
Business Telegraph Code.

CIRCULATION STATEMENT

Of this issue, 36,000 copies are printed. None sent free regularly, no returns from news companies, no back numbers. Figures are live, net circulation.

Contents

Page

Wachusett Dam Hydro-electric Plant 774
Power Plant of the Charlton Mills.. 777
A Power Plant Data Sheet..... 779
Steam and Hydro-electric Power Plant 781
An Emergency Rheostat..... 783
Simple Synchronizing Method..... 783
Generator Trouble..... 784
What Reversed the Motor..... 784
Blown Fuse..... 784
Operating Gas Engine..... 785
The Junkers Engine—V..... 786
Cost of Exhaust Steam Heating.... 788
Editorials 793-794
Readers with Something to Say..795-797
Pump Valve Deck Badly Cracked ....Feed Pipe Heavily Scaled.... Catching the Oil....Silencing Noisy Grab-hooks....Heated Crankpin Indicator....Substitute for Vacuum Cleaner....Chain Sling for Removing Cylinder Heads....Measuring Offset Distance Between 45-deg. Fittings....Test of Cumberland Coal and Screenings Mixed....An Ingenious Measuring Device.... Soapsuds Lubricator for Air Compressor....Beaded vs. Unbeaded Tubes....Emergency Repair for Stripped Thread Bolt.
Questions Before the House.....798-800
Shorter Cutoff, Greater Load.... Friendliness and Familiarity.... What Broke the Stop Valve?.... Air Compressor Efficiencies.... Steam Gage Problem....Emergency Repair of Air Compressor.... Mr. Wylde's Troublesome Injector ....Excess Air Indicated by Superheat.
Inquiries of General Interest..... 801
Engineers' Study Course—IV..... 802
Largest Steam Turbines in New England 804
Some Old Engines in New England.. 805
Large Air Compressor Plant at Panama 806

POWER

Wanted, a Name for Steam-Stuff

The English language is deficient in a specific term for H_2O in a mixed state. When it is in the vaporous or gaseous state it is called "steam." If the temperature is above that at which water changes into steam at the existing pressure, it is "superheated steam." If its temperature is that of the boiling point of water under the existing pressure, it is called "dry saturated steam" if the last trace of moisture has been evaporated out of it, or "moist steam" if it carries some entrained moisture. For the liquid and solid forms we have the common terms "water" and "ice."

But when steam is expanded in a cylinder or a turbine it gives up a portion of its heat, and some of it consequently condenses. How shall we speak of the quantity of H_2O, which is now in question, and which is neither steam nor water, but a mixture of the two?

It is common to say, for example, the *steam* after expansion contains so and so many units per pound. What is meant is that each pound of the *mixture* contains the stated amount of heat. That part of it which is steam contains the full amount of heat per pound given in the table for the dry-saturated condition. The mixture is no more "moist steam" than it is "partly evaporated water." That part of it which is evaporated is steam and that part of it which is not evaporated is water; and that part of it which is steam carries just as many heat units per pound as does dry-saturated steam of the given temperature or pressure. What we need is a general comprehensive word which will signify the substance, H_2O, in whatever physical condition, simple or complex, it may be; some simple appropriate term which will satisfy the gropings of authors who hit upon such terms as "mixture," "steam stuff," "working fluid," "medium." etc. Other languages are just as deficient as our own in this respect, and anybody who will suggest a generally acceptable name for this quantity will fill a gap in scientific nomenclature and render an international service to engineering literature.

Cost of Exhaust Steam Heating

An article on other pages of this issue shows how more power can be obtained from the same amount of steam before utilizing it in the heating system. If the practice obtaining in many large plants is any indication, as long as the power furnishes sufficient steam for heating or the heating system utilizes all of the exhaust steam, the owner seems to think that the maximum efficiency is obtained regardless of how wasteful either the power or heating apparatus may be. Heating and power problems are seldom considered in conjunction and correlated in design. The most economical equipment for power alone is frequently installed and also what is thought to be the most economical heating plant (generally from the standpoint of installation cost and operation in zero weather) without much consideration for the economy of the combination in average weather.

To provide for outside weather changes, all heating systems require for economical operation methods of varying the quantity of heat admitted. This may be accomplished by providing automatic heat control and curtailing the time of operation if the temperature of the medium is nearly constant as in a vacuum system, or by varying the temperature of the medium itself as in the case of a hot-water system. The above requirements are independent of the temperature at which the steam is utilized. The lower the temperature the more radiation required and the greater the cost; the higher the temperature the less the radiation and first cost. As long as the steam consumption is varied through the same range the total amount of heat per season will be the same.

The efficiency of the engine or the amount of steam used per unit of power, will vary directly with the difference in

temperature of the steam entering and leaving the engine. Therefore the temperature of the exhaust absolutely fixes the economy of the engine for power, other things being equal. The latent heat is practically the same for all temperatures at which exhaust steam is used for heating, so that nearly the same heat will be obtained per pound of steam whether utilized at a high or low temperature for heating. It is impracticable to use exhaust steam for heating under final temperatures most economical for power, except in very mild weather, due to the excessive cost of installation that would be required. It therefore becomes necessary to decrease the power economy of the engine to obtain steam at temperatures high enough to bring the cost of the heating system within reasonable limits. The amount of the sacrifice in operation is the charge against the heating system.

This will be the difference in steam consumption due to the final temperature most economical for power operation, and that required by the heating system at any outside temperature period. It has nothing to do with the amount of heat required for the heating system, but does affect the steam required for power. This difference in the steam rate will apply to every kilowatt-hour taken from the engine, whether the resultant exhaust steam is utilized or not in the heating system. If the engine plant is modified to provide high temperatures for heating in winter, the economy of the summer power load is reduced accordingly.

In the article it is shown how a vacuo hot-water heating system utilizing exhaust steam between atmospheric pressure and twenty-eight inches of vacuum will allow the maximum power to be obtained from the combination. Vacuum is carried on the engine ninety-five per cent. of the time and as the weather moderates the steam consumption per kilowatt-hour is greatly reduced. The heating system is performing the function of a condenser for the power load furnishing the exhaust steam.

A vacuum steam system requires exhaust steam at atmospheric pressure throughout the entire heating season, which in the case of a turbine would so increase the steam consumption that the saving would be negligible, to say nothing of the reduction of the rated capacity of the machine.

The article in no way contradicts the general question of the saving engendered by utilizing exhaust steam, but shows how the heating can be done more economically without destroying all vacuum more than five per cent. of the time. The question is not so urgent when the power is inadequate to furnish sufficient steam, but becomes vitally important when the peak load occurs and there is a surplus. This occurs in almost every plant at times, so that a close study of the article may be worth while.

Standards That Need Standardizing

In this day of enthusiasm over the standardization of everything possible, strange it is that the engineering profession has so long overlooked the various quantities and units of measurement used in its own work.

Take the British thermal unit. It is defined as the quantity of heat required to raise the temperature of one pound of water one degree Fahrenheit at or near 39 degrees. Again it is defined as above except that the temperature used is 60 degrees. It is also defined as $^1/_{180}$ of the heat necessary to raise one pound of water from 32 to 212 degrees Fahrenheit. While the difference is small, in any exact use of the British thermal unit the particular one used must be defined to prevent misunderstanding.

Our steam tables are numerous and their variations are also numerous. The various computers have devised formulas representing more or less accurately the results of Regnault's experiments, and all three of the above definitions of the British thermal unit are used. In any exact use of the steam table it is necessary to mention the particular table used and in acceptance tests it is possible to have a test either above or below the guarantee, depending on the tables used in working up the data.

A horsepower is defined as the energy required to raise 33,000 pounds one foot per minute against the attraction of gravity. But as the attraction of gravity varies with latitude and altitude to the extent of ½ of 1 per cent., it is necessary to include the latitude and altitude in an exact definition of horsepower.

The Centennial rating of a boiler horsepower was the evaporation of 30 pounds of water, fed at 100 degrees Fahrenheit, into steam at 70 pounds per square inch. The American Society of Mechanical Engineers has adopted a boiler horsepower as the equivalent evaporation of 34.5 pounds of water per hour from and at 212 degrees. This differs from the Centennial rating only by about 5 British thermal units. Due, however, to the variation of the British thermal units and the steam tables the actual energy represented by a boiler horsepower on the basis of the present definition may vary considerably.

Practically all rating of steam engines is in pounds of water per indicated horsepower. Unless the pressure and quality of the steam are given, these ratings mean nothing. How much simpler it would be if steam engines were rated in British thermal units per indicated horsepower. Then it would be possible to compare the ratings of various engines on a more equal basis. Also in very accurate tests the rating of a steam engine would be affected by the variation in the value of the horsepower, as mentioned above.

Gebhardt, in his book on power plants, makes the following statements in regard to viscosity.

"Viscosity may be defined as the degree of fluidity or internal friction of an oil. It is sometimes called the 'body.' It is determined by a viscosimeter. There are a number of different instruments for this purpose but no recognized standard instrument or method, so that 'viscosity' conveys no meaning unless the name of the instrument, the temperature, and the amount of oil tested are given." It is hardly necessary to comment on the need for a standard in the measurement of viscosity.

Probably other examples of standards that need standardizing might be mentioned, but the above are sufficient to show the desirability of so defining them that it is not necessary to tack an appendix on every report, giving the definitions of the quantities used.

It is good to hear of one with so abounding a faith in his fellows as William B. McKinley, president of the Illinois Traction Co. who has loaned $50,000 in fifteen years to needy students.

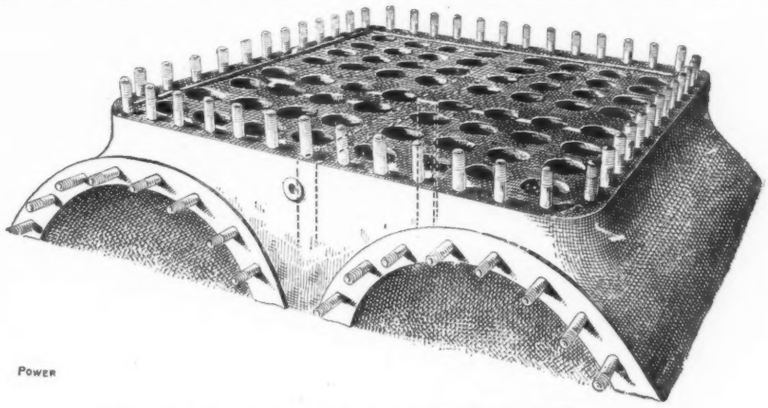
Readers with Something to Say

A letter good enough to print will be paid for. Ideas, not mere words, wanted

Pump Valve Deck Badly Cracked

The illustration shows the discharge valve deck of a 16 and 24 by 13 by 18-in. duplex pump running under a discharge pressure of 120 lb. There are 128 four and one-half-inch valves; more than enough to take care of the water displaced by the plunger.

I would like to have interested readers give their opinions as to what cracked the valve deck as shown. The casting is 4½ ft. across the top and the crack



SHOWING CRACKED VALVE DECK AND PARTITION WALL OF PUMP

extends nearly the full width, being open ⅛ in. at the surface. The thick division wall in the middle of the casting is also cracked down to the suction-valve deck. The side walls or other parts of the pump show no signs of fracture or flaws. The metal is quite heavy, being from 1½ to 2 in. thick.

This is the second casting made from this pattern and both have broken in the same place. Is the break due to carelessness in molding, to an inferior metal or to expansion and contraction?

JOHN MCGINNESS.

New York City.

Feed Pipe Heavily Scaled

The two large injectors and feed pump gave trouble in supplying water to our 60-in. by 16-ft. horizontal tubular boiler. After renewing the disks in all globe and check valves and still being troubled, I decided to disconnect the feed pipe throughout its entire length to locate the obstruction, as I concluded such was the cause of the trouble.

The 1½-in. feed pipe inside the boiler has nine distributing outlets through ½ x6-in. nipples. These were so coated with scale that the openings were slightly more than $\frac{1}{16}$ in. in diameter. It was necessary to renew all the internal piping, after which the pump and injectors worked splendidly.

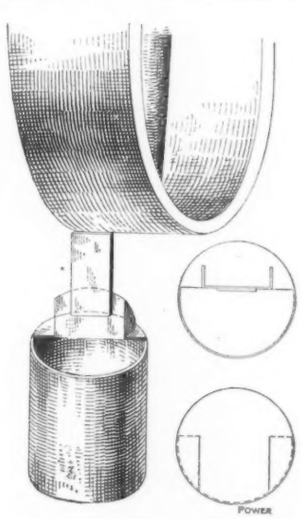
A. N. CARPENTER.

Albany, Ga.

Catching the Oil

Most engineers have been annoyed at times by oil working out on the shaft from the bearings to the hub of the flywheel and on the rim, and from there, due to centrifugal force, being thrown on the walls, floor and ceiling.

The best way to remedy this would be to prevent the oil from getting on the



CAN FOR CATCHING OIL

wheel, but sometimes the flywheel sets so close to the bearing that nothing will prevent oil reaching it, except reducing the supply until there is no excess, which is somewhat dangerous.

A simple remedy is to cut the top from a tin can, as shown by the dotted lines in the illustration, bend it as shown by the heavy lines, solder it in this position and solder a strip of tin to the lid. The can is then to be set in such a position that the tin strip will press lightly against the face of the wheel, the rim running against the end of the tin.

Such oil as reaches the wheel rim will be wiped off into the can instead of decorating the engine-room floor and wall. A large nut or other heavy article in the bottom of the can will serve to hold it steady.

If the appearance of the can is objectionable the wiper can be put on a neat pan which will answer the purpose as well.

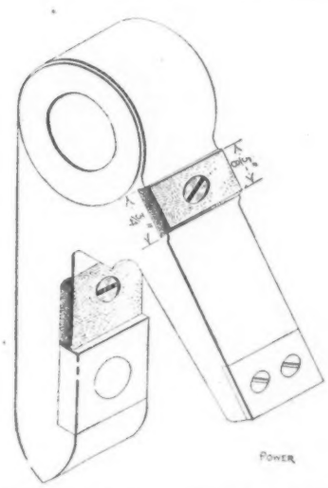
EARL PAGETT.

Coffeyville, Kan.

Silencing Noisy Grab-Hooks

For those troubled with noisy grab-hooks on Corliss valve-gear, I submit the following remedy which I have used successfully.

Where the back of the hook strikes the



GRAB-HOOK FITTED WITH LEATHERS TO PREVENT NOISY OPERATION

spring post when released by the cam, cut a dovetailed slot $\frac{3}{16}$ in. deep by ⅝ in. wide and undercut $\frac{1}{16}$ in. on each side. In the center of the slot drill and tap for a $\frac{3}{8}$x½-in. machine screw. Cut a piece of heavy sole leather to fit the slot accurately, and in this drill a hole that will be directly over the hole in the hook, when the leather is in place.

The hole in the leather should be countersunk so that the head of the screw will be about ⅛ in. below the surface of the leather. After the screw has been tightened, place the hook on its pin, and before the spring is in place, dress the leather down with a file until the plates engage to the required depth. The cushion leather on the other arm of the hook should be used as usual. The illustrations clearly show how the work should be done.

Do not wet the leather, as it will shrink when it dries, work the screw loose and cause trouble.

V. C. EASON.

Mapimi, Dgo., Mexi-

Heated Crankpin Indicator

On all the large stationary engines under my charge, I use the following telltale to indicate a hot crankpin:

On the inside and back of the oil guard, or casing around the crank disk, I have enameled the surface a pure white color, when the pin gets hot the color of the lubricating oil changes, and when it is thrown against the white surface, the engineer can immediately tell that the color of the oil has changed from normal and can at once take steps to prevent a seriously hot crankpin or shut down the engine if it becomes necessary.

A. GORDMAN.

Swissvale, Penn.

Corrosion of Crosshead Bore

Will some interested reader give his opinion regarding the cause of the following trouble?

Less than two years ago the threads of the piston rod and crosshead of our engine were found to be eaten away. A new rod with a larger thread was made and the crosshead rebored. The same thing has happened again and I am desirous of finding the cause. The corrosion is most pronounced in the middle of the threaded section of the crosshead.

The threaded end of the crosshead is made like a split nut with the two parts about ⅛ in. apart, which would allow oil and steam to blow from the packing sleeve into it.

This engine ran for about 16 years before this trouble developed and it seems as though acid in the oil must cause the trouble.

H. K. WILSON.

New Bedford, Mass.

Substitute for Vacuum Cleaner

We have a substitute for a vacuum cleaner that others may have at hand and not realize. As we have no air compressor to clean the generators, we had to wipe them out the best we could by hand. One day a small piece of the packing in the flange joint, which connects our draft tube to the fan-wheel case blew out. We tried to plug it with white lead, but the suction was so great it would draw it through. We finally stopped it with a piece of shingle.

We decided we could use that suction to draw the dirt out of the generators, so drilled and tapped a hole in the tube for a 1-in. pipe, put on a 1-in. gate valve and reduced down to ¾ in. so as to use a ¾-in. hose. It is surprising the way it will pick up the dirt. The tube is 18 in. in diameter, 15 ft. long and water sealed. All we bought were two 1-in. nipples, one 1-in. gate valve, one 1-in. to ¾-in. reducer and the hose. It will pick up pieces of waste, or most anything that will go through a ¾-in. opening. The generators are 25 ft. from the hose attachment at the gate valve.

V. C. WOOD.

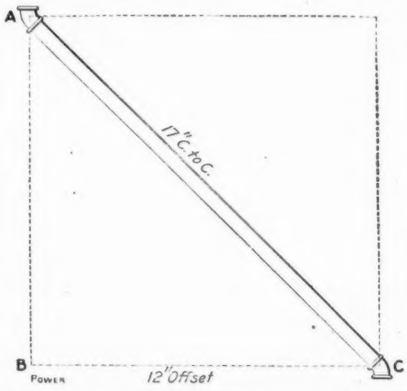
Copenhagen, N. Y.

Measuring Offset Distances Between 45-deg. Fittings

The rule given herewith is one that I find a great help. It is unnecessary with this method to hold the pipes in position or to lay the fittings on a bench, after one has learned this rule. Anyone who knows the multiplication tables can understand the explanation. The rule is an old one but not so generally understood as it should be:

Rule: Muliply the actual offset in inches by 17 and divide the product by 12; subtract whatever is necessary for the fittings and the result is the correct length required in inches.

Cut the pipe that length. If the distance from *C* to *B* is exactly 12 in. and



ILLUSTRATING MEASUREMENT OF OFFSET DISTANCES BETWEEN 45-DEG. ELLS

the 45-deg. ells are placed as shown in the drawing, the distance from *A* to *C* must be 17 in., center to center, of the 45-deg. ells, and that is where the 17 comes from in the rule.

The offset is to be understood as the difference in position, from center to center, of any two desired runs of pipe. In the drawing there is a 12-in. difference, or a 12-in. offset. The length of the pipe required to properly make the desired offset with two 45-deg. ells would be 17 in. from center to center of the ells "made up." Take out for the two ells and the remaining distance is the exact length to cut the pipe.

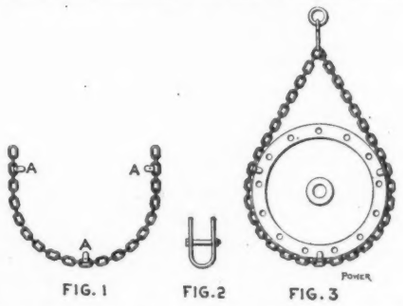
EUGENE OSTER.

Cincinnati, Ohio.

Chain Sling for Removing Cylinder Heads

The illustrations show a chain sling I have used with success for removing cylinder heads.

Fig. 1 shows the sling when ready to put around the edge of the cylinder head.



CHAIN SLING AND CLIPS FOR REMOVING CYLINDER HEADS

Fig. 2 is a piece of flat iron bent U-shape and fastened to a link of the chain with a bolt. Three such pieces are used as shown at *AAA*, Fig. 1.

To use this sling, the nuts are taken off the studs and then the head is pryed forward sufficient to allow the three U-shaped pieces to slip over the edge of the cylinder head. Then by proper tension on the sling, the head can be taken off easily without injury to the threads of the stud bolts. Fig. 3 shows the application of the sling.

JAMES MCCLURE.

Fletcher, N. C.

Test of Cumberland Coal and Screenings Mixed

The following gives the results of a test made in my plant to determine what mixture of the above fuels is the most economical. Clean unmixed Cumberland was first tried; then ⅔ Cumberland with ⅓ hard-coal screenings, and then ⅓ Cumberland with ⅔ screenings. All possible data were recorded. The plant consisted of three 72-in. horizontal return-tubular boilers, one 400-hp. Corliss double-eccentric engine and two other engines of 40 hp. each. The economy effected by screenings mixed with Cumberland coal is a much disputed subject. Some claim there is no economy in mixing the coal, while others use nothing else but ½ to ⅓ screenings mixed with Cumberland coal.

The Cumberland coal used for these

tests cost $3.25 per short ton; the screenings $2. The weight of water evaporated per dollar's worth of fuel is 6485 lb. for pure Cumberland; 6110 lb. for ⅔ Cumberland and ⅓ screenings; 6738 lb. for ⅔ screenings and ⅓ Cumberland.

From my experience I favor about ⅓ screenings mixed with ⅔ Cumberland.

I once had charge of a plant that had two 72-in. boilers burning four tons of coal every ten hours, and I did my own firing with straight coal. I suggested to my superintendent that if he would let me have a fireman, and burn the Cumberland mixed with ⅓ screenings that I could pay the fireman good wages and give my services to repair work. This was done, making things better all

TEST RESULTS ON BOILER BURNING MIXED CUMBERLAND COAL AND SCREENINGS

	Cumberland	⅔ Cumberland ⅓ screenings	⅓ Cumberland ⅔ screenings
Weather	Oct. 4 fair	Oct. 5 rain	Oct. 6 fair
Time	hr. min 11—16	hr. min 11—20	hr. min. 11—25
Boiler pressure, aver. lb.	76.72	75.42	76.18
Feedwater temperature, aver. deg.F.	205	206	207
Tank water temperature, aver. deg.F.	168	177	179
Coal fired, lb.	4770	5950	6553
Moisture-total lb.	73	60 100	33 222
Ash and refuse, lb.	570	788	1024
Ash, per cent.	11.1	12.6	14.9
Combustible, lb.	4359	5223	5510
Steam, quality per cent	0.98	0.98	0.98
Water evaporated total lb.	48,776	51,881	51,828
Water evaporated per lb. coal, lb.	9.89	8.63	7.93
Water actually evaporated corrected for priming, lb.	48,190	51,260	51 215
Equivalent evaporation into dry steam from and at 212F., lb.	50,260	53,360	53,300
Equivalent evaporation per lb. coal, lb	10.30	8.88	8.16
Equivalent evaporation per lb. combustible	11.53	10.45	9.67
Coal burned per hr. per sq.ft. grate, lb.	6.87	8.34	9.14
Damp. flue gas, aver. deg.F.	336	332	357
Draft, natural, inches.	¼		

around, giving me more time for plant upkeep, which would have been impossible had I used mixed coal, because of the increased work of mixing and handling the extra ashes.

A. C. WALDRON.

Revere, Mass.

An Ingenious Measuring Device

Often in a drawing it is necessary to divide a line into three, four or more equal parts. Unless the distance to be divided is a number of units divisible by the requisite number of divisions the operation is anything but an exact one ordinarily. To overcome this all that is necessary is a small rubber band of uniform width and thickness. Suppose that the distance is $4\frac{4}{16}$ in. and it is required to divide it into ten equal parts.

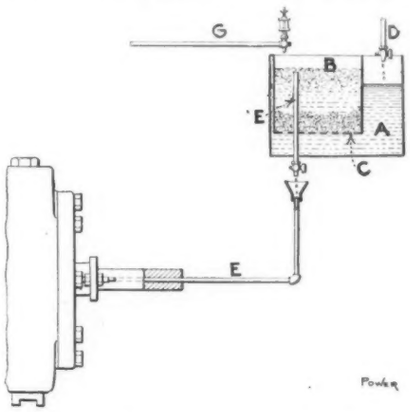
Take a rubber band and mark on it ten spaces of a quarter inch each. Now stretch the rubber until one part of the scale is on one side of the distance to be divided and the other part of the scale is on the other end. Now point off on the piece to be divided, the several stretched divisions as indicated by the rubber. The rubber stretches uniformly, so each division stretches the same until each one is exactly one-tenth of the space between.

A. L. HODGES.

Charleston, S. C.

Soapsuds Lubricator for Air Compressor

The illustration shows a soapsuds lubricator for feeding soapsuds to an air compressor compressing air to 90 or 100 lb. gage pressure in one stage. The temperature of the air after compression is frequently very high, which results in carbon being deposited on the discharge valves and passages, and trouble is therefore experienced in properly lubricating the air cylinder, especially where compressors are run at high speeds and the temperature of the intake air exceeds 80 deg., or where an inferior grade of oil is used. To overcome this trouble, soap-



SOAPSUDS LUBRICATOR FOR AIR COMPRESSOR

suds have frequently been used as a lubricant with good results.

The lubricator consists of a small galvanized-iron water tank *A*, about 7x6x12 in. Inside of it a smaller one *B* is soldered to the large one with its bottom 2 in. above the bottom of the large tank. In the bottom of the small tank are a number of ⅛-in. holes at *C*. Soft brown soap, cut into small pieces, is put into tank *B*. Water is then fed into the large tank through a ¼-in. pipe *D*, and passing through the small holes at *C* dissolves the soap and rises to the top of the ¼-in. pipe *E*, through which it passes down into the inlet of the compressor. The water is regulated by the valve in pipe *D*, the small cock under the tank being left wide open.

Just before shutting down the compressor the water is turned off at *D*, and oil is fed into the small tank *B* from a cup *G*, and the compressor is run with oil for a half hour to prevent rust formation. Soap is used that will sink in the water; otherwise small chips will float and pass down through the pipe *E*. This lubricator is used on a 14x18-in. compressor, and when running at 150 r.p.m. with an air pressure of 90 lb., the water is fed at about 45 drops a minute.

MARTIN MCGERRY.

Holley, N. Y.

Beaded vs Unbeaded Tubes

As this is a subject frequently discussed by engineers, I beg leave to offer a word on it. The beaded end is to be preferred to the unbeaded, though I am satisfied that the latter is more immune from burning off, etc., probably explained from the fact that the structure of the metal is undisturbed. ereas in the other instance it becomes weakened.

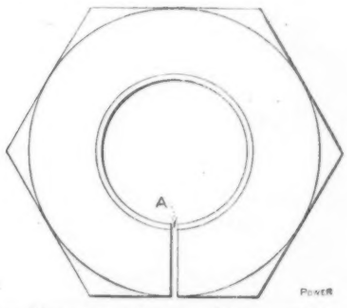
As I see it, all the value that can be attached to the beaded over the unbeaded end is that the bead stiffens the end very materially, and when it is once set up hard and tight it is not so likely to spring away from the head, due to contraction and expansion. If it was left unbeaded there would be more likelihood of its leaking, as sometimes happens when the ends receive a too severe blast of cold air.

LLOYD V. BEETS.

Nashville, Tenn.

Emergency Repair for Stripped Thread Bolt

Sometimes the thread on a bolt or a nut will get stripped; no other is on hand, and a temporary job must be done in a few minutes.



SPLIT NUT FOR USE ON BOLTS WITH STRIPPED THREADS

With a hacksaw cut a slit in the nut as at *A*. If it is a large nut, a few heavy blows with a hammer will bring it together. A small nut can be brought together in a vise. It is surprising what a strain can be placed on the nut without forcing it open at the cut.

JAMES E. NOBLE.

London Junction, Ont., Canada.

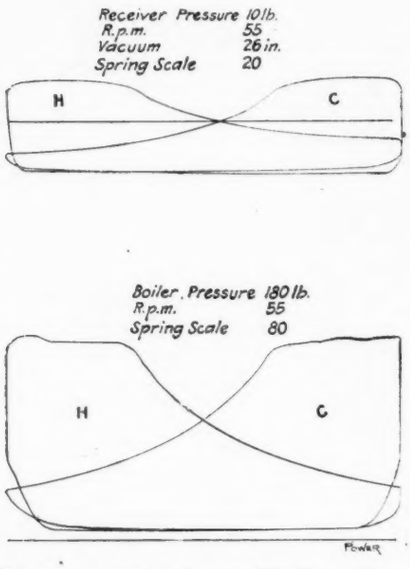
Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues

Shorter Cutoff, Greater Load

This subject, as POWER has treated it, has greatly interested me. As I am in no way proficient on the subject I will not offer an argument, but I wish some interested reader would criticise the two indicator diagrams which were taken from a 14 and 28 by 42-in. tandem compound Corliss engine, directly connected to a 70,000-gal. rotary pump.

These cards were taken while the engine and pump were working against an 8-ft. head, but when we have to raise the water against a 9-ft. head, the engine will not carry the load unless I lengthen the cutoff on the low-pressure cylinder, thus reducing the receiver pressure from



DIAGRAMS FROM 14 AND 28X42-IN. TANDEM-COMPOUND CORLISS ENGINE

10 to 6 or 7 lb. according to the load. I have occasionally tried manipulating the cutoff on the low-pressure cylinder to raise the receiver pressure to 18 or 25 lb., which I have no trouble in doing, but in this case the high-pressure valves do not trip regularly, but drag, thus making the engine race so badly as to raise and lower the governor so that the safety cams operate, stopping the engine.

We have another unit of the same size and working under the same conditions, which will carry the load at 8 or 9 ft. head and 14 lb. receiver pressure without any trouble. I would be pleased to hear from someone who could tell me how to remedy the trouble without raising the boiler pressure. Surely it can be done, as this engine has always carried the load with ease the same as the other one does, at 14 lb. receiver pressure, but for some reason it will not now.

THEODORE PLACETTE.

Rosedale, Tex.

Friendliness and Familiarity

I read the editorials, "Know your Employees" (July 16) and "Friendliness and Familiarity" (Sept. 24) with pleasure, as they are in accord with my practice, that is, get close to the men. The editorial did not appeal to me as it apparently did to Mr. Pierce.

All of us are men and should be treated as such. Simply because one is chief engineer, is it necessary or right for him to keep constantly before the men the fact that he is chief, and to impress upon them that they are subordinates, inferiors, etc.? The fireman may perhaps be a college graduate doing such work to learn the practical side of the business. In this case one would be "entertaining angels unawares." Such actions toward men, seem to me like the very essence of snobbishness.

It is said that "familiarity breeds contempt"; it may, but I have never yet found the familiarity as it exists in the engine and boiler room breeding contempt for the chief, if he is a true man. But I have found a universal contempt for a *martinet* (an arrogant disciplinarian). I have found it a good thing to get as close to those under me as conditions will allow, make them feel that they are an important part of the plant. By taking this course the men become interested in their work. More intelligent work can be had of them by this means, than by "building a fence" between by acts of snobbishness.

I want the men's ideas. Many good ideas have been obtained from the fireman and even the coal passer. I know that discipline must be maintained, but to do so it is better to make the men feel that they are men worthy of recognition. As the great Napoleon let his men know that they were constantly under his eye and that he noted their bravery, so let the men under us know that their work is constantly observed and the good work duly appreciated.

Mr. Pierce says: "When a manager, superintendent or even chief engineer gets the habit of being familiar or allowing the men under him to become familiar, then the wheels of good management start to clog, and in time a new crew, or new head man will become necessary." This is not so unless the manager is utterly ignorant of the characteristics of men.

As an example of snobbery and "pigheadedness," take the following:

A young man, chief engineer of a large plant, insisted that the men, every morning, when reporting for duty should touch their caps and say "good morning chief," and when anyone was present, address him as chief. This young man did not last long, even the management could not tolerate such a snob.

F. C. B. SPEARE.

Cape May City, N. J

What Broke the Stop Valve?

On page 463 of the Sept. 24 issue, Mr. Keil asks the opinion of POWER readers as to what caused a stop valve to break. I believe it was due to water-hammer. I do not think that closing the valve on boiler No. 1 had anything to do with the accident to the valve on No. 2 boiler.

With the drain pipe connected to the boiler-feed line, as shown in Mr. Keil's sketch, I cannot see how the water could drain from the pipe between the valve and the header, as there is more pressure on the feed line, due to friction than on the boiler.

Mr. Keil states that the pressure on all the boilers was equal at the time of the accident. That being so, the instant valve No. 2 was opened the flow of water was in the direction of boiler No. 2, because the pressure in the pipe, due to the head of water, was greater than that in No. 2 boiler. The water flowing toward No. 2 might therefore have caused a damaging water-hammer against the valve. If the pressure on No. 2 had been enough greater than that in the header to cause the steam to flow into the header when the valve on No. 2 was opened, I do not think the accident would have occurred, as there would have been no cause for water-hammer.

When cutting a boiler into a header, it is well to have the pressure on the boiler a little higher than that on the header.

S. F. FARLEY.

Galveston, Tex.

Air Compressor Efficiencies

In the article by E. M. Ivens in the Oct. 15 issue it is stated that the mechanical efficiency of an air compressor should include the horsepower equivalent of the heat put into the jacket water added to the indicated horsepower of the air cylinder. In the first place, it may be mathematically shown that all of the power of an air compressor goes into heat. After air has been compressed and then cooled back to its original temperature at constant pressure, it contains no more energy than it did before. It is possible, by figuring air weights and temperatures due to compression, to show that the air horsepower exactly equals the heat appearing in the air, if it is not cooled. If, now, the air is partly cooled by the jackets, a portion of this heat goes into the water jacket; but the total heat, represented by the temperature and weight of air and temperature and weight of water, is still equal to the actual though reduced indicated horsepower, barring radiation of course. Carrying this still further, if the jackets could perfectly cool the air during compression there would still be the indicated horsepower of isothermal compression and then there would be all the horsepower in the jacket water, which under these theoretical circumstances would be equal to the isothermal horsepower itself. This would give twice the isothermal horsepower as the actual power developed, which would result in some delightfully high mechanical efficiencies something like 110 or 120 per cent., allowing for the heat still left in the air and the mechanical friction.

To say that the horsepower in the heat of the water jackets should be added to the indicated horsepower of the air cylinder is as outrageous as to say in a prony brake test that the horsepower of the cooling water should be added to that calculated by the pressure of the brake arm. The heat is the result of the work done and equal to it in both cases, so that their sum would give twice the horsepower actually developed.

In the third column on page 579, referring to Fig. 2, it is stated that the observed volumetric efficiency is $ad \div bd$. In the first place this gives another case of 110 per cent. efficiency, because *ad* is longer than *bd;* but even if this is written backward, it would not be any better because the true volumetric efficiency as shown by the indicator diagram is $bc \div ad$. Here *bc* represents the actual amount of air admitted to the cylinder, at atmospheric pressure. The length *bd* would not represent anything real, unless to show the clearance loss only. The length *dc* is as clearly a loss as *ab* and sometimes, with some kinds of inlet valves, *dc* far exceeds *ab*.

To blame the compressor for the kind of air furnished it, as Mr. Ivens does, when he says volumetric efficiency depends upon temperature and pressure of intake, is like upbraiding a man for eating poor food when he is too poor to buy a better quality. The volumetric efficiency of a compressor is the ratio of its actual output in cubic feet referred back to the temperature and pressure of the actual air furnished to it. This is just as logical as to credit a steam engine with the heat passing away in the condensate. If anybody defined, as Mr. Ivens claims, the volumetric efficiency as referring to 14.7 lb. absolute pressure and 60 deg. F., the definer took a good deal upon himself, for one might say such atmospheric conditions occur about once a year at some places, and never at all at others.

As a matter of fact, altitude would really affect a single-stage compressor somewhat, because a given gage pressure at an altitude means a greater number of compressions than at sea level and so a greater clearance loss by expansion. A compound machine would be affected hardly at all, because the number of compressions in the first stage, where the volumetric efficiency is produced, depends upon the cylinder ratio, which in a given machine is fixed. The ratio of compression in the high-pressure cylinder would be slightly greater, due to altitude, and so its efficiency would be slightly less, tending to increase the effective cylinder ratio, and so slightly reducing the efficiency of the low-pressure cylinder. This effect, however, is small and entirely unrelated to Mr. Ivens' remarks, as he simply charges the compressor because the atmosphere is below 14.7 absolute at high altitude. Obviously this is unfair.

As to heating of the air through the compressor-intake passages, this is chargeable against the machine because of the design, but this heating is impossible to measure, after the air is actually in the cylinder, so it does not apply to the elaborate formulas appearing in the article in question.

By the way, what is the use of burdening an article with such formulas, even if they do happen to be right? Several of the expressions given could be more easily solved by simply measuring the card lengths and dividing one by the other. The proceedings of the American Society of Mechanical Engineers had a staggering collection of just such useless formulas on gas compression not long ago, many of which were simply ratios of card lengths.

I have used the system of measuring air, as depicted in Fig. 6, quite extensively, but it is not so accurate as it at first appears. The trouble is to obtain the final, or even the initial, temperature of the air in tank *C*. Owing to the pressure, a thermometer well is necessary, and there is considerable lag in temperature rise. I have used a thermocouple, but have some doubts of this also. If a cooling coil is inserted between tanks *B* and *C*, the air going to tank *C* is certainly cool and this temperature is not hard to measure; but some able pneumatic engineers claim that the air in *C* will become heated again due to the compression of the increasing pressure within itself. This looks to me like perpetual motion, because this would be adding heat and so energy, to tank *C*, and there is no place for this to come from except the motor driving the compressor. This second tank *C* should then increase the power required to drive the machine, but it does not because the power of the motor must be the same, with or without *C*, as long as the regulating valve keeps the discharge pressure in *B* constant. The energy cannot come from air in *B* because its pressure and temperature remain constant, and if the heat is removed by a cooling coil between *B* and *C*, the energy put into it by the compressor motor is expended in this cooling coil. I rather think that this two-tank method, if a cooling coil is used between the two tanks is a good one and commercially accurate.

After all his mathematical discussion, Mr. Ivens asks the use of the term "volumetric efficiency" if it depends only upon clearance. There is no use for it if it is figured his way, because he has entirely neglected the loss at the "toe" of the card. This latter item is a matter of inlet-valve throttling, producing a pressure below atmosphere at the commencement of the stroke. With proper mechanical valves and suitable intake, this pressure may equal atmosphere or actually exceed it a little, when air inertia is allowed to act, just as in the setting of inlet valves for modern automobile engines.

Besides all this, however, Mr. Ivens has said but little about slip, except in scored cylinders and with leaky valves. There is as much, and more, to learn about this item as there was, and still is, to learn about the discrepancy between steam consumption by the indicator diagram and by the condenser.

R. S. BAYARD.

New York City.

Steam Gage Problem

In answer to the inquiry of Angelo Belmonte regarding his defective steam gage, I would say that possibly the spring has lost its elasticity. Then, again, I think he will find that the small gears that mesh to produce the movement of the hand have become so worn that their movement is impeded. I would suggest that he examine the gage again to ascertain if this is not the reason the gage registers a wrong pressure.

I once knew of a steam gage that would work all right up to 100 lb. pressure, but the hand would stay there un-

til the pressure rose to 110 lb. when it would jump from the 100-lb. to the 110-lb. mark. As the safety valve opened at 110 lb. the fireman was somewhat surprised when the gage first acted this way, although it had been working badly for some time. It would register 100 lb. and then the pointer would gradually move on until 110 lb. was registered.

The gage was taken apart and it was found that where the gears meshed the constant vibration of the pointer had caused a shoulder to be worn on two of the teeth. This shoulder caused the gears to bind at 100 lb., but when the pressure rose to 110 lb. it was enough to overcome the resistance offered by the shoulders. The shoulders were filed off and the teeth smoothed up and when the gage was put back it worked all right.

LEON L. POLLARD.

Fairfield, Maine.

Mr. Wylde's Troublesome Injector

Relative to E. S. Wylde's troublesome injector as mentioned in the Sept. 24 issue, I offer the following: As the injector itself has been tested and found all right, it is logical to suppose that something in the manner of connecting it up is wrong. In the first place, the steam supply to the injector should not be taken from the header, for unless the header is of ample capacity, the flow of steam from it to the engines or other apparatus will be detrimental to the proper working of the injector. Instead, the injector steam-supply pipe should be connected directly to each boiler through a system of valves and pipes, so that steam can be taken from any boiler, and that the other boilers may be shut off from the injector at will.

Another feature of the connections as illustrated in Mr. Wylde's letter which I believe to be bad, is connecting the injector discharge pipe into the pump discharge pipe through a tee. If the injector must discharge into the pump discharge, then it should do so through a Y-fitting. Possibly these two changes may cause the injector to work as it should. If not, then the suction pipe should be increased to 2 in. diameter and the discharge pipe made one size larger than it is at present.

CHARLES J. MASON.

Scranton, Penn.

Reading about E. S. Wylde's injector trouble reminded me of a similar one I was called upon to remedy.

The plant was a small one, just erected, and the injector refused to work. I noticed that the piping was all too small, and after replacing the piping with that of the proper size, I tried the injector, and found to my surprise that it would not work. I next examined it and found the steam nozzle out of line with the combining tube. I took it back to the dealer, who argued that it was all right, until I showed him the trouble. He gave me another of the same make, which has been doing good work ever since, and that was about four years ago.

The injector acted exactly like Mr. Wylde's, that is, it would prime, but not force. I suppose it could not get velocity enough to open the check valve against the pressure, so when the tubes were out of line, the water all came out of the overflow.

PERRY BARTRAM.

Cedarpoint, Ill.

I have recently had trouble with an injector similar to that mentioned by E. S. Wylde. After trying repeatedly I was unable to get water into the boiler with this injector, so I started the feed pump, but without success. I then took the injector apart and found it almost choked with small fragments of packing from the city main pump, these I removed. After replacing the injector it furnished water for the boiler without further trouble.

I informed a friend of this remedy, who had me look over his plant to see why his injector, which had recently been purchased, would not supply the boiler with water under the pressure for which it was recommended. I took the injector apart and found it so heavily scaled that the proper amount of steam could not enter the nozzle. When cleaned this injector worked satisfactorily.

The manager then asked me to look over a new injector in a small lighting plant in a neighboring town. This injector never worked and they had to run cold water into the boiler with a feed pump, or let the steam run down to 30 lb., and put the lights out. Upon examining it I found it in good condition. I then decided to trace the feed line to the boiler, and later entered the boiler and found a cold chisel in the mouth of the injector discharge line so badly scaled that it almost closed the opening. When the boiler was steamed up again the injector was tried and it worked well. If Mr. Wylde will see that the delivery tube is clean, the check valves in working order, the feed pipe to the boiler clear, the injector nozzles clear and repeat some of the foregoing processes, he will remove his difficulty.

J. E. STROTHER.

Yuma, Ariz.

Emergency Repair of Air Compressor

Mr. Bayard's communication on page 689 of the Nov. 5 issue has been read with much interest. The writer's original description of this emergency repair job in the issue of Sept. 17 was, perhaps, not so complete as might have been desired.

The cross-compound compressor referred to was a 20 and 8 and 12 by 18-in. and not an 8-in. stroke machine. The 10x10-in. compressor was a duplex machine.

The high-pressure cylinder of the cross-compound compressor having been wrecked in starting up, by water coming from a split intercooler tube, and the 10x10-in. duplex machine being of insufficient capacity to give the required amount of air for the operation of the plant, the discharge from the low-pressure cylinder of the cross-compound compressor was piped into the suction of the duplex machine.

The only reason for limiting the pressure on the suction of the duplex machine to 25 lb. was, that it was not considered advisable to carry a pressure higher than 25 lb. in the suction base or bedplate in which the suction chamber was comprehended.

The arrangement as outlined, supplied sufficient air for the requirements of the plant, and apparently did not strain the small compressor, and prevented a costly shutdown; which was the end sought.

Mr. Bayard's analysis of the problem is unusually interesting and instructive, but, on a breakdown job, there is little time for elaborate analysis or calculation. The important thing is to "get going."

W. M. FLEMING.

Holyoke, Mass.

Excess Air Indicated by Superheat

On page 268 of the Aug. 20 issue the editorial on "Excess Air Indicated by Superheat" is interesting. While excess air causes a reduction of furnace temperature, reducing the amount of heat transmitted to the water in the first pass, it is not apparent why the temperature of the gases should not be reduced to the same point through the first pass, as if the furnace temperature was higher, unless the velocity of the gases, on account of their greater volume, would be increased so much that insufficient time would be available for heat transfer.

Would it not be reasonable to suppose or assume that the increase in superheat with larger amounts of air is due to the fact that excess air, reducing and wasting furnace temperatures, reduces the capacity of the boiler, and therefore the amount of steam passing through the superheater is less? With the velocity reduced and with the same transfer of heat, the temperature of the lesser steam volume would be greater.

It would be interesting to learn if the evaporation at times of high superheat and excess air was less than when opposite conditions prevailed.

SOREN ANDERSON.

Butte, Mont.

Inquiries of General Interest

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it

Piston Valve

What advantage has the piston valve over the common D-slide valve?

E. D.

The piston valve is balanced and can be operated with very much less friction than the ordinary slide valve, and most forms have the advantage of being easier to construct and repair.

Flash Point of Oils

What is meant by the flash point of an oil?

E. D.

The flash point is the temperature at which an oil gives off inflammable gases. Oils whose specific gravities are below 0.85 generally have flash points below 60 deg. F. while those whose specific gravities exceed 0.85 usually have flash points above 60 deg. F.

Closing Cracks in Boiler Settings

What is a good method of stopping cracks in the walls of boiler settings?

J. H.

Fill the cracks to a depth of 2 or 3 in. with asbestos wool, leaving a crevice at the surface of the wall about ¼ in. deep and fill with a putty made of pulverized lime and linseed oil having a small proportion of asbestos wool mixed through the putty for a binder. The elasticity of the asbestos material and plasticity of the putty will cause those materials to remain much longer in position to keep the cracks sealed than a pointing of ordinary lime or cement mortar, which after setting, soon becomes crumbled from expansion and contraction of the brickwork.

Volume of Air per Pound of Coal

In ordinary steam-power plants, what volume of air is required for combustion per pound of coal?

N. P.

Theoretically about 12 lb. of air per pound of coal is required, varying according to the proportions of the combustible ingredients. With natural draft and ordinary furnaces, about 24 lb. of air is required per pound of coal to secure perfect combustion; and with artificial blast and some kinds of coal the amount of air may be only a small percentage in excess of the theoretical amount required for complete combustion.

A cubic foot of air at 60 deg. F. and under atmospheric pressure at sea level, weighs about 536 gr. and 13.06 cu.ft. weigh 1 lb. Therefore under the atmospheric conditions stated the necessary air for complete combustion of a pound of coal is about $12 \times 13.06 = 156.72$ cu.ft., but with ordinary furnaces and natural draft about twice that quantity, i.e., about 313 cu.ft. is usually required, and with artificial blast from about 160 to about 300 cu.ft. of air per pound of coal.

Distribution of Draft in Flues

How may the uniformity of distribution of furnace gases among the fire tubes of a return-tubular boiler be determined?

M. F.

Considering the products of combustion to be of uniform temperature at their entrance to the flues at the rear tube sheet, the relative temperatures of gases emerging from the flues may be regarded as indicative both of relative draft and relative efficiencies of the flues. The temperatures of emerging gases may be determined approximately by placing in the tubes near their ends small pieces of substances of known temperatures of fusion as follows:

Spermaceti	120 deg. F.
White wax	154 deg. F.
Sulphur	239 deg. F.
Tin	449 deg. F.
Bismuth	518 deg. F.
Lead	621 deg. F.
Zinc	786 deg. F.
Antimony	1166 deg. F.
Aluminum	1218 deg. F.

Low Boiler Pressure vs. Reduced High Pressure

Is it more economical to operate a low-pressure steam-heating plant by generating steam at the pressure required for use, or to carry a higher boiler pressure and use the steam reduced to the same low pressure through a pressure-reducing valve?

A. B.

The amount of heat delivered by the boiler to the point where the reducing valve discharges would be the same in either case. But in carrying higher boiler pressure more heat would be radiated from the boiler and the piping up to the reducing valve, as their temperatures would be higher. Hence if the additional heat so radiated were not utilized there would be loss of economy on this account, due to carrying higher boiler pressure. An additional loss would arise if carrying higher pressure required using a boiler-feed pump, and there also would be some loss with higher boiler pressure due to a falling off of boiler efficiency on account of less efficient absorption of furnace heat by a higher boiler temperature.

Oil Fuel vs. Coal

What are the advantages and disadvantages of petroleum as fuel compared with coal?

H. A. J.

The advantages of petroleum are: (1) Much lower cost for handling, dispensing with cost of stoking and removing ashes. (2) Less space occupied for equal heat value, and storage possible at a considerable distance if desired. (3) Instant regulation of the intensity of combustion to conform to the demand for steam. (4) Increased boiler efficiency and capacity and greater elimination of smoke by keeping furnace doors closed. (5) Less labor required around a plant. (6) Comparatively no dust, dirt or smoke. The principal disadvantages are: (1) Serious conditions imposed by insurance or municipal regulations respecting location of oil tanks, and (2) increase in boiler-repair bill from overheating and burning of boilers by formation of "steam pockets" due to intense heat acting on boiler parts not protected by rapid circulation.

Exhaust Steam Returns

Will low-pressure heating coils, fitted with automatic air valves and supplied with exhaust steam, be more effective in warming a building, if the returns are collected in a closed receiver, or if the returns are run dry to the receiver and the latter is provided with a vapor connection left open to the atmosphere?

W. H. N.

A vapor outlet from the receiver to the atmosphere would make the coils more effective as it would better insure removal of air and a more positive circulation of steam in the heating system. The vapor connection should be provided, however, with a stop valve for partly closing off the escape as might be found necessary to prevent a high velocity and short-circuiting of steam in the mains past the heating surfaces.

Engineers' Study Course—IV

Decimal Fractions

In engineering calculations involving fractions the problems are usually worked out in decimal fractions, because they are more easily handled than vulgar fractions.

We know that one figure as 1, standing alone is one—is a unit. If we add one cipher to the right as 10, we increase the value of 1 ten times; another cipher added would increase the value of 10 ten times or to 100, etc. Every cipher or *place* as it is called added to a given figure increases the value of the preceding figure or figures by 10.

1. 10. 100. 1000. 10,000

Notice that the ciphers are at the left of the period (.), and that every cipher added at the left increased the value of the figure by 10.

Now suppose we carry out this operation still further by turning these figures end for end as,

1000.
100.
10.
1.
0.1
0.01
0.001
0.0001

Just as we increased the value by 10 for every cipher put between the 1 and the period, we decrease its value by 10 for every cipher put between the period and the 1. The period, by the way, is called a *decimal point*. The 1 at the right of the decimal point then is a *tenth* because its value is only $\frac{1}{10}$ that of one, and the expression 0.1 is one-tenth. Continuing to read down the column we say,

0.1 = *tenth*
0.01 = *hundredth*
0.001 = *thousandth*
0.0001 = *ten-thousandth*

There are two chief things to remember about decimals. One is that the number of places at the right of the decimal point equals *the number of ciphers in the denominator of the same fraction expressed vulgarly*, as

$0.1 = \frac{1}{10}$ = 1 *tenth*
$0.01 = \frac{1}{100}$ = 1 *hundreth*
$0.001 = \frac{1}{1000}$ = 1 *thousandth*
$0.0001 = \frac{1}{10000}$ = 1 *ten-thousandth*

The other is that the decimal point *affects only the numbers at the right of it*, for any number at the left is a whole number.

Thus,

$8.01 = 8\frac{1}{100}$, $266.266 = 266\frac{266}{1000}$

Whole numbers and decimals as, 2.5, 3.05 and 4.25 are mixed numbers the same as though expressed $2\frac{5}{10}$, $3\frac{5}{100}$ and $4\frac{25}{100}$.

Inserting a cipher between the decimal point and the decimal divides the decimal by 10.

$0.3 = \frac{3}{10}$; $\frac{3}{10} \div 10 = \frac{3}{100} = 0.03$

The opposite of this is true if we take away a cipher from between the decimal point and the decimal; that is, the decimal will be multiplied by 10.

The value of a decimal is not changed by adding or suppressing ciphers at the right of the decimal.

$0.5 = \frac{5}{10}$; $0.50 = \frac{50}{100}$ and $\frac{5}{10} = \frac{50}{100}$

It is plain the value is the same because

$0.5 = \frac{5}{10} = \frac{1}{2} = 0.5$ and $0.50 = \frac{50}{100} = \frac{1}{2} = 0.5$

Thus you will know whenever you see the expression 0.50 that it does not equal any more nor less than 0.5. This applies to all like expressions.

But, *putting a cipher between a decimal and a decimal point divides the decimal by* 10.

Using the same figures as above and inserting a cipher as mentioned.

$0.5 = \frac{5}{10}$; $0.05 = \frac{5}{100}$

We learned that the denominators of decimals multiplied themselves by 10 for each decimal place so,

$0.5 = \frac{5}{10}$; $\frac{5}{10} \times \frac{1}{10} = \frac{5}{100} = 0.05$

The opposite of this is true if we take a cipher from between the decimal point and the decimal, that is,

Suppressing a cipher between a decimal point and a decimal multiplies the decimal by 10.

Using the same figures as above,

$0.05 = \frac{5}{100}$; $\frac{5}{100} \times 10 = \frac{5}{10} = 0.5$

Addition of Decimals

In addition of decimals, whole numbers *must* be kept at the *left* and decimals at the *right* of the decimal point.

Therefore the only difference between adding whole numbers and decimals is the position of the decimal point.

Whole numbers	Decimals	Mixed numbers
6226	0.6226	4.6226
81	0.81	10.81
948	0.948	2.948
7255	2.3806	18.3806

Notice the decimal point continues down in a straight line and that the decimal is close up to the point.

Example: The following number of gallons of oil were used in one month: 2, 11.86, 5.65, 6.225. How many gallons were used altogether?

2.0
11.86
5.65
6.225
25.735 gal.

Rule: *Place the numbers so that the decimal points are directly under one another. Add as with whole numbers, putting the decimal point in the sum under the points above it.*

Subtraction of Decimals

The position of the decimal points and decimals is the same for subtraction as for addition of decimals.

Example: 0.8864 — 0.252

Minuend 0.8864
Subtrahend 0.252
Difference 0.6344

Example: A bunker contained 667.22 tons of coal; during the week 18.5, 26.56, 30.1 and 50.25 tons were used, how many tons remained?

18.5 + 26.56 + 30.1 + 50.25 = 125.41 *tons used*

then

667.22
125.41
541.81 tons remained

Example: A tank contained 22 gal. of oil; 0.225 gal. were drawn out. How many gallons remained?

22.000
0.225
21.775 gal. remained

Rule: *Place the figures and decimal points as in addition of decimals and subtract as with whole numbers, putting the decimal point in the remainder directly under the points in the minuend and subtrahend.*

Multiplication of Decimals

No attention is paid to the position of the decimal point in multiplication of decimals *until* the *product* is found. We place the numbers as in multiplication of whole numbers and proceed to multiply as in multiplication of whole numbers. Then point off as many places in the *product* as there are decimal places in the multiplicand and multiplier combined.

Example: 0.25 × 0.60

0.25
0.60
0.1500

There are four decimal places in all, so there must be four places in the product.

Example: A boiler furnace burned 1125.25 lb. of coal per hr. How much was burned in 4 hr.?

1125.25
4
4501.00 lb.

There are only two decimal places required in the product because the multiplicand and multiplier combined have only two places. The two ciphers represent zero or nothing so they may be dropped.

Example: Multiply 0.464 by 0.002

$$\begin{array}{r} 0.464 \\ 0.002 \\ \hline 0.000928 \end{array}$$

In this example we get only 928 after multiplying by the 2. We cannot put the decimal before the 9 because there are six decimal places in the multiplicand and multiplier combined. To get six places we prefix enough ciphers to the 9 to give the decimal places called for. Rule: *Put the multiplier under the multiplicand. Do not consider the decimal points. Multiply as with whole numbers and point off as many decimal places in the product as there are places in the multiplicand and multiplier; if necessary, prefix ciphers.*

Division of Decimals

In placing the figures in division of decimals we pay no attention to the decimal points until the quotient is found. But, *we must have as many decimal places in the dividend as there are decimal places in the divisor, even though we add ciphers to obtain them.* Then divide as in whole numbers. Now to obtain the correct number of decimal places in the quotient, we must *subtract the number of decimal places in the divisor from the number of places in the dividend; the number of places remaining is the correct number of places to point off in the quotient.*

Example: 0.625 is to be divided by 25.

$$\begin{array}{rl} \text{divisor dividend} & \text{quotient} \\ 25 \;)\; 0.625 & (\; 0.025 \text{ ans.} \\ 50 & \\ \hline 125 & \\ 125 & \end{array}$$

There were 3 decimal places in the dividend and none in the divisor, consequently one cipher must be prefixed to give the correct answer.

Example: A boiler evaporated 805.18 lb. of water with 93.5 lb. of coal. What was the evaporation per pound of coal?

$$\begin{array}{rl} 93.5 \;)\; 805.180 & (\; 8.61+\text{lb.} \\ 748\;0 & \\ \hline 57\;18 & \\ 56\;10 & \\ \hline 1080 & \\ 935 & \\ \hline 145 & \end{array}$$

In this example there are three decimal places in the dividend and one in the divisor, a difference of 2, so we point off 2 places in the quotient. It will have been noticed that a cipher was added at the right of the dividend. This, as we learned, does not change the value of the fraction, and in this case it was done that it might be added to the 108 remainder so as to carry the fraction out further than the original dividend allowed. We could have carried out the division to any number of decimal places desired by adding a cipher for each place. In carrying out the division in this way point off when all the original figures in the dividend have been used. After doing this we need pay no further attention to the original dividend. When it is not desired to carry the decimal any further than 2, 3 or 4 places, if it does not come out complete, adding a + sign to the quotient indicates that the decimal could have been carried further.

When the divisor contains more decimal places than the dividend, add enough ciphers to the dividend to make them equal.

Example: Divide 10 by 0.0005.

There are four decimal places in the divisor and none in the dividend, so by adding to make the number of decimal places equal in the divisor and dividend we have:

$$0.0005 \;)\; 10.0000$$
$$20000$$

We cannot subtract any decimal places from the dividend because the number of places in both the divisor and dividend are equal; therefore there is no remainder and no decimal places are needed in the quotient.

Reduction of Fractions to Decimals and Vice Versa

Example: Express ½ decimally.

$$\begin{array}{r} 2 \;)\; 1.0 \\ \hline 0.5 \end{array} \quad 0.5 = \tfrac{1}{2} \text{ ans.}$$

Example: Express ¾ decimally.

$$\begin{array}{r} 4 \;)\; 3.00 \\ \hline 0.75 \end{array} \quad 0.75 = \tfrac{3}{4} \text{ ans.}$$

Rule: *Add ciphers to the numerator and divide by the denominator. Point off as many decimal places as there are ciphers added.*

To reduce a decimal to a fraction.

Example: $0.75 = \frac{75}{100} = \frac{3}{4}$

Rule: *Under the decimal, place a one with as many ciphers as there are figures in the decimal, then reduce to lower terms.*

Suppose we wanted to express 0.573 as a fraction having any given denominator, say 64. Then

$$0.576 \times \tfrac{64}{64} = \frac{36.864}{64} \text{ or approximately } \tfrac{37}{64} \text{ ans.}$$

Rule: *Multiply the decimal by a proper fraction having the desired denominator, and the same numerator so that the value of the fraction is unity.*

Examples for Practice

1. Express decimally, one-tenth; nine-hundredths; ninety-five-thousandths; three and nineteen-hundred and fifteen ten-thousands; one hundred and twenty-three and nine-hundredths.

2. Add together, 1010.24; 2.222; 19.005; 0.0026; 51.2; 3.1416.

3. The total weight of water pumped into a boiler was 1029.56 lb. The pump was stopped and 350.05 lb. of the water was evaporated. How much remained?

4. A pump delivers 4.59 gal. per stroke and makes 42 strokes per minute. How many gallons would be pumped in 219.5 min.?

5. An engine showed a steam consumption of 27.5 lb. per hp. per hr. If the engine consumed 15,125 lb. in one hour and 9.5 lb. of steam were generated per pound of coal, how many horsepower did the engine develop and how many pounds of coal were used per horsepower per hour?

Answers to Last Week's Questions

(1) $2409\frac{4}{25}$ lb.
(2) $20{,}390\frac{3}{8}$ lb.
(3) $2790\frac{55}{1365}$ lb.
(4) 4.
(5) 100.
(6) 24.

NEW PUBLICATION

FARM GAS ENGINES. By H. R. Brate. Published by the Gas Engine Publishing Co., Cincinnati, 1912. Size, 5x7 in.; 195 pages, illustrated, cloth. Price, $1.

The large number of gasoline engines now employed for driving tractors, pumps, etc., as well as farm-lighting plants, seemed to warrant the publishing of a book devoted to the special needs of the farmer in this line. With this in view, the text has been prepared to cover in an elementary manner the principles of the gas engine, its operation and its care; since most of the common troubles may be traced to ignorance or indifference on the part of the operator.

The author has followed the usual custon of beginning the book with an early history of the gas engine. The next two chapters are devoted to explanations of the two-stroke and four-stroke cycles. Chapters 4 and 5 deal with fuels and carburetion and in this connection considerable space is devoted to the use of kerosene in gasoline engines. Ignition, lubrication and cooling are next taken up, followed by a very practical chapter on farm-lighting plants. The book concludes with a list of the common gas-engine troubles with directions for locating and remedying them.

While written primarily for the use of the farmer, the suggestions apply equally to marine, automobile and all medium-sized stationary gas and gasoline engines.

BOOKS RECEIVED

MODERN ORGANIZATION. By Charles DeLano Hine. "The Engineering Magazine," New York. Cloth; 110 pages, 5x7½ in.; indexed.

COMMERCIAL ENGINEERING. By Alfred J. Liversedge. Emmott & Co., Manchester, England. Cloth; 369 pages, 5½x9 in.; indexed. Price, 7/6.

HOW TO READ A DRAWING. By Vincent C. Getty. J. B. Lippincott Co., Philadelphia, Penn. Cloth; 64 pages, 6x9¾ in.; fully illustrated. Price, $1.

Largest Steam Turbines in New England

There are now being operated in the New England states, five steam turbines, each of 15,000 kw. capacity, a total of 75,000 kw. These are the largest in New England. Two of these machines are operated by the Edison Electric Illuminating Co., two by the Boston Elevated Co., both of Boston, Mass., and one by the Rhode Island Co., Providence, R. I. These are illustrated in Figs. 1, 2 and 3.

These units are of the same design and a description of one will answer for all. The turbines are six-stage Curtis machines. The four in Boston have Worthington base, surface condensers, each having approximately 22,500 sq.ft. of cooling surface. The centrifugal, volute circulating pumps are each driven by a vertical Atlantic steam engine at a variable speed of from 150 to 200 r.p.m. The speed depends upon the temperature of the cooling water, which varies from about 30 to 70 deg. F.

These condensers are connected to a

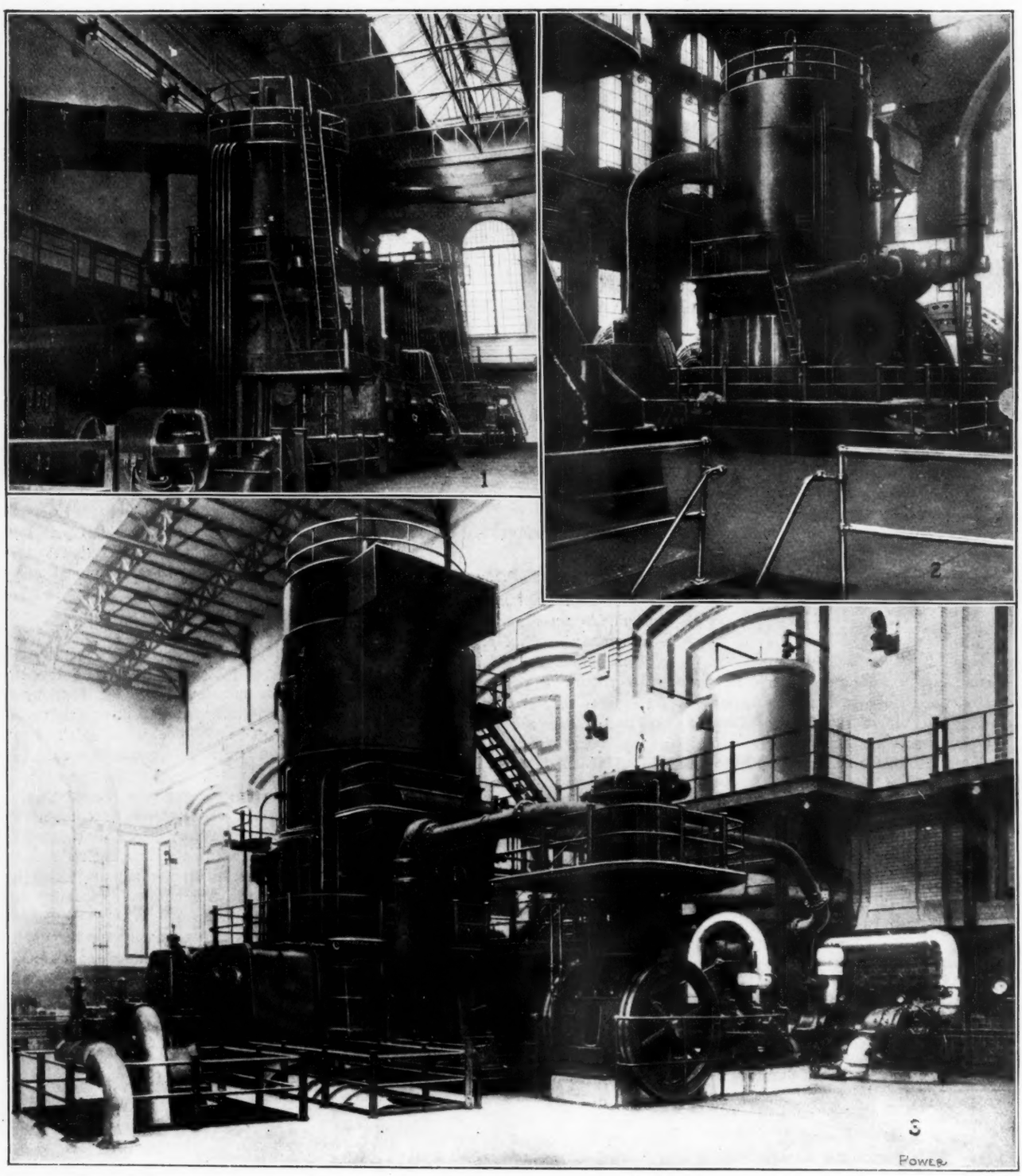


FIG. 1. TWO 15,000-KW. TURBINES IN THE PLANT OF THE BOSTON ELEVATED CO. FIG. 2. A 15,000-KW. TURBINE UNIT OF THE RHODE ISLAND CO. FIG. 3. ONE OF THE TWO 15,000-KW. TURBINES OF THE BOSTON EDISON ILLUMINATING CO.

12 and 31 by 24-in. combination horizontal and vertical Laidlaw-Dunn-Gordon single-cylinder reciprocating dry-vacuum pump and a centrifugal 5-in. wet-vacuum pump directly connected to a 26-hp. Curtis turbine. Water to the step bearings is delivered by a 14 and 3½ by 18-in. duplex pump. A Goulds triplex gear-driven pump takes care of the oiling system, and is driven by a 7½-hp., 550-volt induction motor.

Each turbine is directly connected to a 10-pole, 7000-volt, revolving-field, 60-cycle, three-phase General Electric alternator.

The Providence turbine runs at 750 r.p.m. and is directly connected to an 11,000-volt, 788-ampere, three-phase, 60-cycle alternating-current generator. It is connected to a No. 20 double Leblanc condenser in the basement. The two water-circulating centrifugal pumps are driven by the same turbine coupled to one end of the shaft. This turbine has taken over the entire load of the station that was formerly carried by cross-compound reciprocating units.

Some Old Engines in New England

Several old engines, each having an interesting history, are still being operated and doing good work. Some of them have been in use over 50 years.

New England has been the acknowledged textile manufacturing center of the United States since "long before the war." Scattered here and there may still be found in operation many steam engines that have driven the machinery in individual manufacturing plants since their incorporation, some of them having over 50 years of service to their credit. In some instances all record as to age has been lost, but the fact that they are still in operation demonstrates how well and of what good material these engines of earlier days were built.

Two of these old-time engines are operated every working day at the works of O. B. North & Co., New Haven, Conn. One is a 12x36-in. Corliss, running at 57 r.p.m., shown in Fig. 1. It develops 45 hp. with 80-lb. steam pressure. The engine shaft carries a 9-ft. flywheel, belted to the factory lineshaft.

This engine has an interesting history. It is one of the first George H. Corliss engines built by the Foundry & Machine Co., Taunton, Mass. It carries nameplate No. 53 and was built under the patent of 1851. Originally it was installed at a lumber mill, where it passed through a fire undamaged. It was purchased by the present owner in 1865 from the original owner for $50. Ten years ago the engine passed through a second fire, and was then removed to a vacant lot, where it stayed for about one year, during which period the brasses and other easily removable parts were detached by covetous junk men.

It was then repaired at a cost of $300, missing parts being replaced and worn ones renewed. Although it is known to have been in operation for more than 50 years, the engine runs noncondensing as smoothly as many newer and costlier machines.

In the same engine room is a 12x36-in. Greene engine, Fig. 2, built by the Pacific Iron Works, Bridgeport, Conn., in 1855. It runs at 65 r.p.m. and carries an average load of 28 hp. The frame is of the solid-box type, with the crosshead guides bolted to the crank side of the frame.

There are two eccentrics. The exhaust valves are operated by a rock shaft that extends through the frame to the exhaust valve-gear under the cylinder. The tappet steam valves are operated by a fixed eccentric; the point of cutoff is controlled by a governor supported by a bracket clamped to the steam pipe about 6 ft. above the steam chest. The governor is driven by a round belt from a groove pulley on the engine shaft. The cast-iron throttle valve is made square with a brass seat and disk. The engine has been in operation 57 years and is run every working day.

The Greene engine shown in Fig. 3 is of practically the same design as that in operation at the North company's plant. It was built by the Providence Steam Engine Co., Providence, R. I., has an 18x48-in. cylinder, and has developed 438 hp. with 125 lb. steam pressure, running condensing at 102 r.p.m. Although installed in 1870 it has been in charge of but three men during its 42 years of

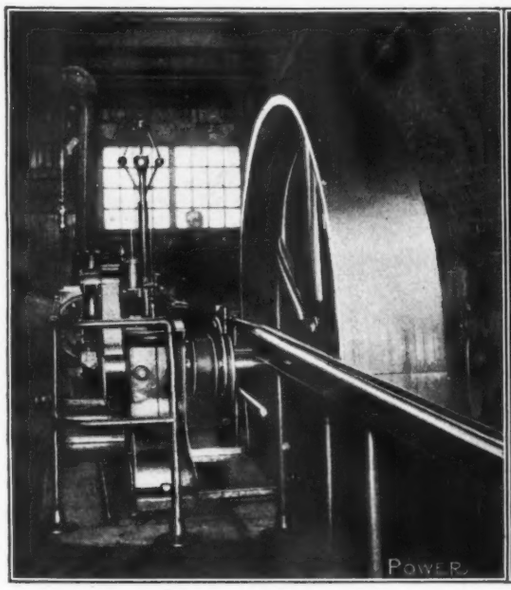


FIG. 1. OLD GEORGE H. CORLISS ENGINE

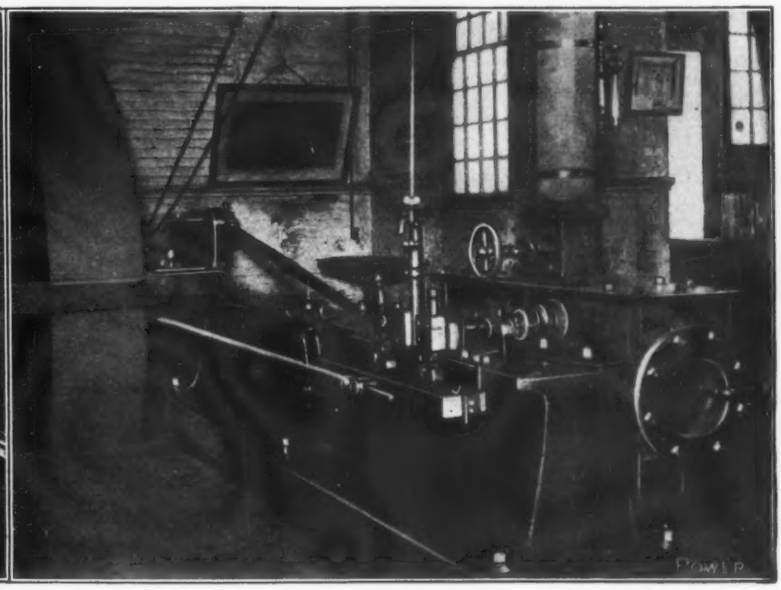


FIG. 2. OLD GREEN ENGINE

service. Luther B. Miller is the present engineer. The governor is steadied by weights on a weight yoke and by a spring dashpot.

The original speed of the engine was 55 r.p.m. This was increased to 67 in 1890 and to 102 r.p.m. in 1900. The original steam pressure was 85 lb. In 1900 the engine was developing 150 hp. Now, 12 years later, the average load is 300 hp., running condensing. This engine is operated by the Dexter Yarn Co., Pawtucket, R. I.

Another old engine with a history, but not now used, is still in the old engine room of the George H. Gilbert Co., Ware, Mass. It is of 50 hp. capacity, and was the first steam engine to be operated in that town. It was built in 1854, or 58 years ago, but has not been in service since 1880, or 22 years, although the driving belt was left in place up to within a year ago. It was built by Corliss & Nightingale, Providence, R. I. The crankshaft is cast iron, and the valves are closed by iron weights. The cylinder is 12x42 in. and the engine ran at 45 r.p.m. It was used to drive the original mill of 48 looms and employing 80 hands. There are now 476 looms and 1600 hands are employed. Unfortunately, the photograph taken was so defective that it is impossible to illustrate the engine.

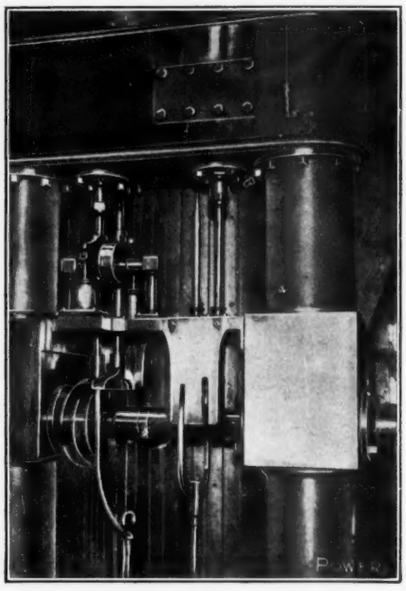


FIG. 4. VALVE GEAR OF BEAM ENGINE

In Fig. 4 is shown the steam side of an old engine running at the New Haven Copper Co., Seymour, Conn. It is of the walking-beam type, has a 42-in. by 7-ft. cylinder and runs at 22 r.p.m. It was built by Woodruff & Beach, Hartford, Conn., in either 1854 or 1855, or 58 years ago. It is constructed much after the design of most walking-beam engines, with an A-frame to support the walking-beam, a crankshaft supported by a solid-box bed on one side of the center frame and a vertical cylinder on the other. The governor hangs from an arm secured to the side of the cylinder.

The engine has poppet steam and exhaust valves, which are operated by cams. The cutoff of the steam valves is controlled by a cone-shaped sleeve, which is moved in or out in a horizontal position by the governor. The latch blocks and cam are oscillated to a front and rear position by gravity as the valve is tripped or latched for opening. The steam valves are at the top and bottom of the cylinder in separate steam chests. Live steam passes through cast-iron upright columns to the cylinder.

This engine runs condensing, a jet condenser being operated by a rod running from the walking-beam to the condenser pump, in the engine-room basement.

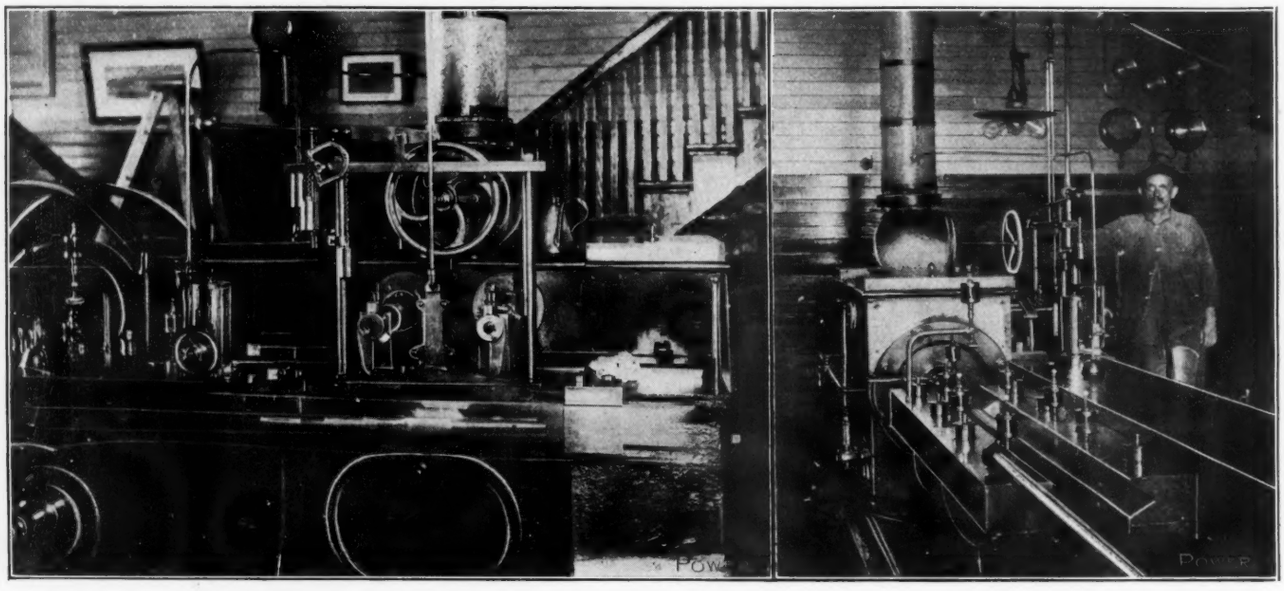


FIG. 3. TWO VIEWS OF ANOTHER OLD GREEN ENGINE

Large Air Compressor Plant at Panama

BY FRED H. COLVIN

One of the largest air-compressor plants in the world is that which supplies compressed air for the work on the Panama Canal. Of the four separate stations, the one at Empire is the largest and the one at Balboa the smallest. The other two are at Las Cascadas and Rio Grande.

The air compressors at Las Cascadas, Empire and Rio Grande deliver their entire output into a 10-in. main, 14 miles long, from which it is drawn off by smaller subsidiary mains to the air drills and other machinery at various points along the canal. The great majority of the air is used in and around Culebra Cut.

The plant at Balboa comprises two Laidlaw-Dunn-Gordon compressors, each capable of delivering 2200 cu.ft. of air into a main four miles long. The two plants thus have 18 miles of air-compressor mains, which is believed to be the longest in existence.

The other three plants contain six Laidlaw-Dunn-Gordon compressors of 2500 cu.ft. capacity each and eight Ingersoll-Rand compressors of the same capacity. The Empire plant has an average total monthly output of over 400,000,000 cu.ft. of free air at 70 deg. F. The largest total monthly output for all the plants, according to the last report, was 794,357,-

608 cu.ft., while the average cost of it for the year was 3.24c. per 1000 cu.ft.

The maintenance of the 14 miles of 10-in. main entails a great deal of trouble, due to the changing character of the soil and the constant jarring of the earth from the daily blasts in Culebra Cut. In 1911, 6300 ft. of the main had to be taken up and relaid, due to its proximity to two slides along the banks of the canal. Patrolling these pipe lines has been found advantageous and a good investment in view of the saving effected in the amount of air lost through strained and leaking joints.

A very close account of the cost of compressing the air is kept month by month, the charges being divided into labor and materials for operating the plants, their maintenance, and any extensions which may be necessary.

From oil-fired boilers an evaporation of 14.53 lb. of water per lb. of oil is obtained. Various tests showed that from 157½ to 168 gal. of oil were equivalent to 1 ton of coal.

The lubricating oil is very carefully accounted for, the performance being given in revolutions of the compressors per gallon. The plant at Rio Grande shows 174,000 revolutions per gallon of valve oil, 177,760 revolutions per gallon of engine oil and 461,065 revolutions per gallon of air cylinder oil, while at the Empire plant the last item is 836,875, or nearly double. The output in compressed air per barrel of fuel oil runs remarkably uniform, the lowest being 42,180 cu.ft. of free air per barrel of fuel oil at the Rio Grande plant and 43,196 cu.ft. for the plant at Las Cascadas.

New Power Development for Connecticut

A dam for power purposes costing $5,000,000 is to be constructed across the Connecticut River at Windsor Locks, Conn., in place of the present dam of the Connecticut River Co. A new $7,000,000 corporation is to be formed, and of this interests of the Connecticut River Co., which owns the present dam and canal, represent about $1,000,000. In addition to the $5,000,000, much more will be expended for an auxiliary steam power plant at tidewater. The new dam will be about 4½ miles below the present dam and is expected to make possible the development of about 35,000 hp.

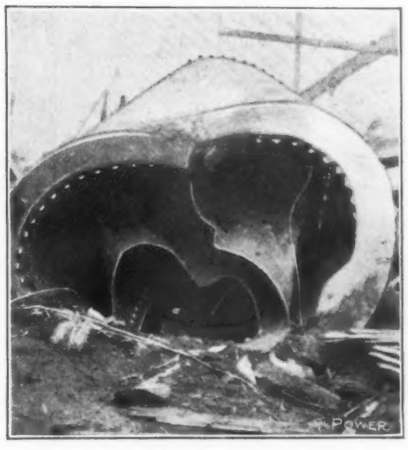
Congress will be asked to grant the franchise to the new corporation, which will probably be called the Northern Connecticut Power Co. That corporation now owns the Enfield Power Co. and supplies electricity for lighting and power from a steam plant. It has the charter right to distribute power to Hartford, Tolland and Litchfield counties and owns also the Connecticut Cable Co., which can operate anywhere in Connecticut.

Webb City Mine Boiler Explosion

Further details of the Webb City, Mo., boiler explosion, Nov. 5 issue, are published herewith.

It was a small vertical boiler and used to run a hoisting engine. It was 30 in. in diameter and 6 ft. high with a double-riveted lap joint. The shell and firebox sheets were ¼ in. thick; there were no stay-bolts, and the fire sheet was riveted at the bottom to a cast-iron mud ring.

The initial rupture was near the mud ring, in the firebox sheet, which was corroded at this point to 1/16 in. thick. There was no safety valve on the boiler, it having been broken off some two weeks previous and the opening plugged. The boiler contained three gages of water ten minutes before the explosion. The steam gage was located at the throttle of the hoisting engine. The boiler was never



EXPLODED BOILER

inspected, and it is probable that the gage was defective.

This is the fourth explosion in this district since the first of the year, in all of which five persons were killed and five badly injured. All of the exploded boilers were old and second-hand, equipped with defective safety apparatus, or none at all.

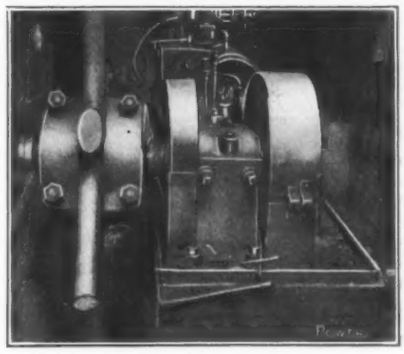
Wauseon Flywheel Explosion

BY OSCAR F. RABBE

An accident occurred at 4:50 a.m., Oct. 31, at the plant of H. S. Williams & Co., Wauseon, Ohio, when a 10-ft. 17-in. flywheel on a Houston, Stanwood & Gamble 10½ and 14½ by 20-in. engine exploded. The wheel weighed about five tons. What was left of it is illustrated herewith. The engine was equipped with a throttling governor.

With the exception of the broken wheel, damage to small piping, walls and roof of the building the loss was small. No one was injured. The wrecked engine was belted to a 60-kw. generator which supplied the commercial and city lighting and at the time was carrying a load of about 25 kw. All lights in circuits were burnt out, due to excessive voltage.

Both engine cylinders were cross-connected so that the engine could be used



EXPLODED FLYWHEEL

as a twin as well as a compound unit. One section of the flywheel rim, weighing about 400 lb., tore out the 4-in. cross pipe, went through a belt hole in the wall, hit the wood-lagged pulley on a generator, and deflecting at an angle of about 30 deg. to the left and upward, struck the ceiling and landed on the floor on top of the cross pipe. A section of the rim, weighing approximately 1360 lb., went through the engine-room roof at an angle of about 40 deg. and landed in a field to the east of the plant some 520 ft. from the engine shaft.

A third piece went through the east and west walls of the company's sawmill, and was buried in the ground. All fractured surfaces were clean, showing no flaws except a blow-hole in a rim section, but to all appearances the wheel failure was due solely to overspeed. The flywheel was of two-piece construction with six spokes; the rim joints were bolted with three 1⅝x9½-in. bolts.

A Helping Hand

That much practical good is being accomplished by the international committee of the Young Men's Christian Association is evidenced by its annual report on its work. During the past year 150,000 foreigners, representing 45 nationalities, have been assisted in reaching their friends, helped to secure situations and taught to speak English. The port secretaries in Europe alone met 802 steamers, gave 54,926 cards of introduction to young men, and wrote 4603 letters for them.

The association's work in teaching trades to foreign as well as native members has been highly indorsed by owners and superintendents of shops and factories, and they are anxious to hire men from the association's trade schools.

Bowser Liquid-Measuring Devices

S. F. Bowser & Co., Inc., Fort Wayne, Ind., have recently brought out three types of pipe-line measuring devices for measuring and recording the quantity of any liquid, other than water.

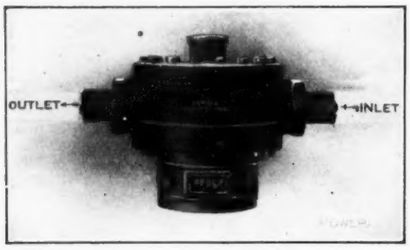


FIG. 1. MEASURE WITH INTEGRATING DIAL

The first type, Fig. 1, measures and records on an integrating dial the number of gallons handled. It is made in sizes for pipes of from ½ to 10 in. in diameter and is applicable in fuel-oil-burning installations for checking up the quantity of oil delivered to the storage tanks, measuring the oil fed to the various burners, etc.

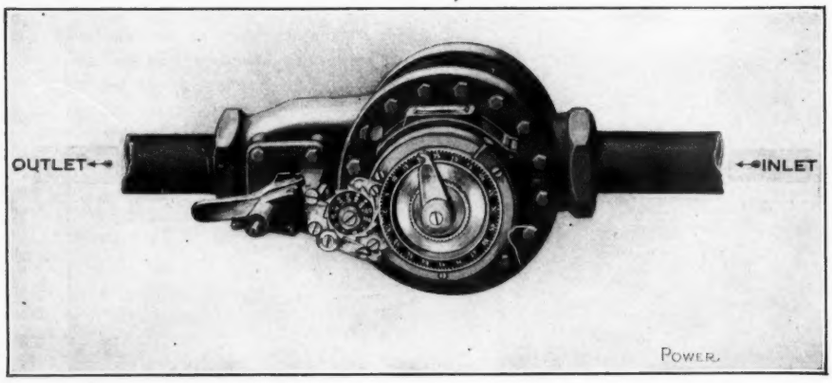


FIG. 2. MEASURE WITH AUTOMATIC STOP

A second type, Fig. 2, is similar in design, except that it has an automatic stop, so arranged that at the option of the operator any predetermined quantity varying from 1 to 1000 gal., can be accurately and rapidly delivered and when this amount has been discharged, the flow is automatically stopped. This device can also be set to duplicate any quantity. When necessary, the measure is steam jacketed to prevent the liquid from freezing.

The remote-control type, Fig. 3, consists of an automatic registering measure and motor-driven rotary pump for handling the liquid. When the predetermined quantity has been delivered, the measure automatically opens the motor circuit and the pump stops instantly. Fig. 4 is a sectional view of the pump showing the principle upon which it is constructed.

An important part of this system is the air separator installed near the measure. It contains a float valve which automatically liberates any air in the pipe and thus prevents it from discharging with the liquid and causing a false reading. This separator contains also a screen of ample area to prevent particles of foreign matter from fouling the measure.

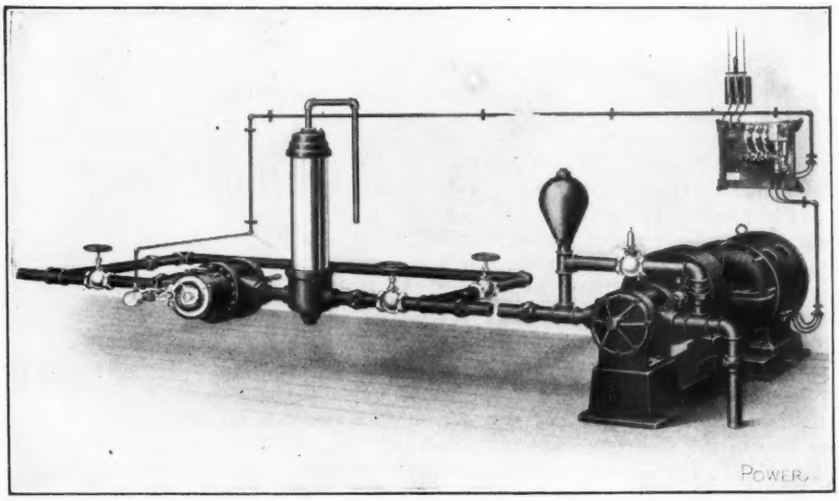


FIG. 3. MOTOR-DRIVEN AUTOMATIC REGISTERING MEASURE

Fatigue of Cast-iron Used with Superheated Steam*

It is now general practice to avoid the use of cast-iron fittings in piping carrying superheated steam, although many such installations are still in use and with no indications of distress in the metal due to the high temperature of the steam.

A case of failure of a cast-iron fitting which formed part of a superheater occurred in the power house of the Union Electric Light & Power Co., of St. Louis. The temperature of the steam in this case was 455 deg. and the temperature of the surrounding gas ran as high as 600 deg. The failure was in the nature of a small crack in a cast-iron outlet, and as there were nearly 5000 similar castings in service under the same conditions, apprehension existed that cast-iron was not a safe metal to use for the purpose. On this account the defective piece was removed and cut up and sample pieces were subjected to careful physical tests. The examination was made in the testing laboratory of Columbia University, and the conclusion reached was that:

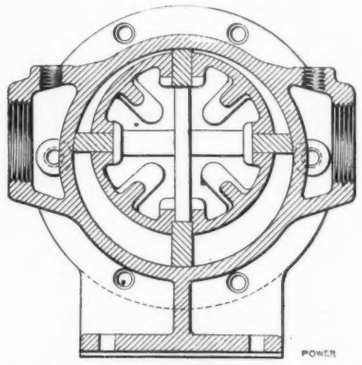


FIG. 4. SECTION THROUGH ROTARY PUMP

> The tensile strength of the cast iron outside the small zone where the crack occurred is quite as high as that of good grades of gray iron, and the condition of service does not seem to have caused a material loss of strength of the metal.
>
> The crack which occurred was due to the segregation of foreign matter in the casting, the area thus affected being small, but in exactly the proper position to receive the stresses which might be induced by the expansion of a pipe in the opening.

Against the earlier experience of the St. Louis company we may instance its later experience and that of various other large companies which were compelled to replace a great number of cast-iron fittings in superheated steam lines because of distortions and failures. A test on pieces cut from the body of a cast-iron valve taken from a superheated steam line of the Commonwealth Edison Co.

*From the report of the Prime Movers Committee of the National Electric Light Association.

was reported in *The Valve World* for March, 1908, and is in part as follows:

The Crane Co. has been very anxious to obtain something definite on this important subject, and the opportunity for investigation presented itself a few months ago when a 14-in. high-pressure gate valve, which had been in service for four years, was taken out and replaced by a steel valve of similar design. The company, having a complete record of the tests of the iron which was used at the time this valve was made, was in a position to determine accurately what the effect of long-continued superheated steam really was. The salient points are:

A loss of strength in the body of 49 per cent., while the metal in the flanges, not being under quite so high a heat and not directly exposed to the action of the steam, showed a loss of only 33⅓ per cent.

Following is the detailed result:

Test on 14-in. No. 9-E cast-iron gate valve, extra heavy, removed from a superheated steam line.

Steam on line, about August, 1903.

Valve taken out, September, 1907.

Time in service, four years.

Pressure of steam, 200 lb. per square inch.

Temperature of steam, about 590 deg., sometimes a little higher.

Original tensile strength of cast iron, as shown by test bars, 22,400 lb.

Tensile strength of bars cut from body of valve, 12,303 lb., 11,608 lb., 10,610 lb., 12,440 lb.; average strength, 11,740 lb.

Loss of strength in body after four years' service as compared with original test bars, 49 per cent.

Tensile strength of bars cut from the flanges on the valve, 14,900 lb., 15,250 lb.; average, 15,075 lb.

Loss of strength, 33⅓ per cent.

The valve above referred to was 22½ in. long when it was installed in the line, and was 22⅝ in. long when it was taken out, showing an elongation of $\frac{5}{16}$ in.

The following extract is from a publication issued by the Crane Co. and entitled "The Effect of Superheated Steam on Valves and Fittings":

Cast-iron subjected to superheated steam having a temperature exceeding 500 deg. F.—and we say 500 deg. simply because we know the effect is produced beyond that point and we do not know whether a lower temperature is equally bad—undergoes a slow, constant increase in volume with a consequent loss of strength.

As steel does not appear to increase permanently in volume at temperatures of 500 to 800 deg. F., we may assume that cast iron is so affected on account of the larger volume of elements, such as carbon and silicon, contained in that metal. It may be noted in this connection that cast iron is made up of approximately 92½ per cent. iron and 7½ per cent. of other elements by weight, but the iron is so much heavier than the carbon, silicon, sulphur or phosphorus that when we consider a cube of cast iron by volume we find the elements other than iron, instead of occupying 7½ per cent. of the cube, take up about one-third, hence there is a very large portion of cast-iron which may be affected. Cast steel, on the other hand, contains only about 0.6 per cent. of carbon, silicon, sulphur and phosphorous, an amount altogether too small to have an effect upon the total volume.

Undoubtedly the deterioration of cast-iron in superheated steam lines is due to frequent and rapid changes in temperature within the range of the superheat. For example, suppose a steam line carrying steam having a normal superheat of 150 deg. If, when running at full load, the greater part of the load were suddenly thrown off, the superheat temperature might rise to 200 or 225 deg. If now the entire load were thrown off, the temperature of the steam and of steam piping would fall 200 deg. or more to the temperature incident to the steam pressure.

Suppose, again, while running under normal conditions and with 150 deg. of superheat, water should be carried over from one of the boilers through the superheater and into the steam line. This water would have a temperature 150 deg. less than that of the steam pipe, and the effect on any cast-iron fitting with which it came in contact would be exactly the same as would be produced by pouring water on a hot stove-plate. The metal would warp and its inner surface would shrink and be permeated with cracks, microscopical in size at first, but which would be extended from time to time until the strength of the entire casting was seriously impaired.

For further investigation of this subject reference may be had to the American Society of Mechanical Engineers' *Transactions* for the year 1910, Vol. XXXI, page 989.

The conclusion is that cast-iron fittings in a superheated steam line constitute a hazard, and that the danger increases with the amount of the superheat, and on this account your committee strongly advises against the use of this metal for such purposes.

The *Travelers Standard* is a new monthly journal published by the engineering and inspection division of the Travelers Insurance Co. and the Travelers Indemnity Co., of Hartford, Conn. It will deal with engineering matters of all kinds, but will be mainly concerned with safety engineering as applied to construction work, manufacturing, mining, power generation and transmission, the electrical and chemical industries, and every other form of activity in which machinery or tools are used, or human life and limb are imperiled in productive work. The company says regarding the policy of the standard: "We shall particularly emphasize practical matters, but we shall also deal with theoretical principles to some extent, because it is impossible to be truly 'practical' without having a better knowledge of a subject than can be gained by merely observing facts and combining them crudely by 'rule-of-thumb.'"

Over the Spillway

Just Jests, Jabs, Joshes and Jumbles

Hen Roost, the popular young fireman at the Pipville featherbed works, says the *Farmer's Advocate*, was some put out on Thanksgiving Day. Hen was corn-feeding a turkey for this festive occasion, but it turkey-trotted for parts unknown the night before. Roost suspicions that the Bulgarians must have slaughtered his "turk" by this time.

Because Thomas Edison, Jr., has declared it his life's ambition to find a substitute for coal, the jokesmiths are poking fun at him. If we had half his enthusiasm—and optimism—we'd drop everything and help him look.

"The Atlanta, Ga., smoke inspector wears a snow-white hat as a smoke detector," says a correspondent. You leave us in grave doubt, sonny, as to the rest of his apparel. Atlanta is a warm town, but we are sure even smoke inspectors must bow to the conventions. Come, now, they must wear something else besides snow-white hats?

Old Doc. Bergonie has discovered that electricity furnishes a means of nourishment for our innards. And just about the time we common peepul have acquired a taste for this kind of diet, some wicked capitalist will have formed a trust in restraint of cheap living, and beat us to it.

It required 200,000 lb. of copper to construct four large brewing kettles of 25,000 gal. capacity each for the Ruppert brewery. Twenty-five thousand gallons of—well, whatever they cook in 'em—is nothing to us, but just think of that large amount of copper!

Sir William Ramsay is said to be working out the details of a "universal primitive sign language." In a manner of speaking, it has Volapuk and Esperanto skinned a mile. We need something of the kind in our operating field, for we have times when no known language seems to have an entirely adequate expression. Some high and primitive sign might turn the trick.

The central station, claims a contemporary, has largely got away from its former "the public be damned" attitude. Some folks are skeptical enough to believe that the central station has frequently got away with more than rightfully belongs to it.

Air Receiver Explodes

By Letson Balliet

On Oct. 30, 1912, the air receiver at the McNamara silver mine, in Tonopah, Nev., exploded, wrecking one end of the compressor room and injuring slightly the night engineer running the hoist. The hoist house stands within 25 ft. of the shaft while on the opposite side of the shaft stands the big mill for treating the ores which are dumped directly from the skips into the bins above the ore breakers. Adjoining the hoist house was a "lean to" shed called the compressor room. The entire structure was of frame covered with corrugated iron. Between the end of the hoist house and compressor room were located two air receivers.

The plant had originally been operated by steam, but had been overhauled and for the last 18 months had been operated by electric power bought from a hydro-electric power company, whose generating plant is in the Sierra Nevada Mountains of California.

A 20¼ and 13¼ by 12-in. Ingersoll-Rand compressor driven by a belt from a 125-hp. motor was uninjured. In May of 1905 a smaller compressor had been installed, with one receiver, and during 1910, the present compressor had been installed, replacing the old one, and a second air receiver connected into the line but the old receiver had been left in use. For a time the steam hoist had been operated by compressed air, but recently the piston rods had been disconnected and a motor had been geared to the intermediate shaft.

The night shift of about 40 men were at work under ground and in the mill, while the hoist engineer was looking after the compressor. The gages had been brought within his vision by removing the partition between the compressor room and the hoist room. The compressor was in sight but the motor was to his right and back of him, while the air receivers were outside the building ahead of the compressor, and between the compressor and the shaft. For some reason the hoist engineer had left his chair, and was about 3 or 4 ft. away from it, when the explosion occurred. In any other spot in the building he would probably have been killed. He was cut by flying débris, the controller for the hoist motor was knocked over and the chair crushed by a piece of the receiver that dropped through the roof. The receiver that exploded was the old one which had been in use since May, 1905. It was 4 ft. in diameter by 16 ft. long and made out of ⅜-in. boiler plate. Longitudinally there were four plates held together by single-riveted lap joints. Both heads were dished and secured with a single row of rivets.

In the explosion the receiver was literally torn in two pieces, similar to the method of opening a can by winding up a strip of the metal on a wire key. A strip 2 ft. wide, badly bent and broken into three or four pieces, was torn out of the middle of the receiver. One head and two tiers of the side plates (about 8 ft. long) was thrown up over the building and dropped through the roof, the strips torn out were thrown in several directions, while one head and about 6 ft. of the receiver moved only a foot or two. The other receiver was blown a distance of 20 ft. and apparently not damaged.

Fig. 1. Scene of the Explosion

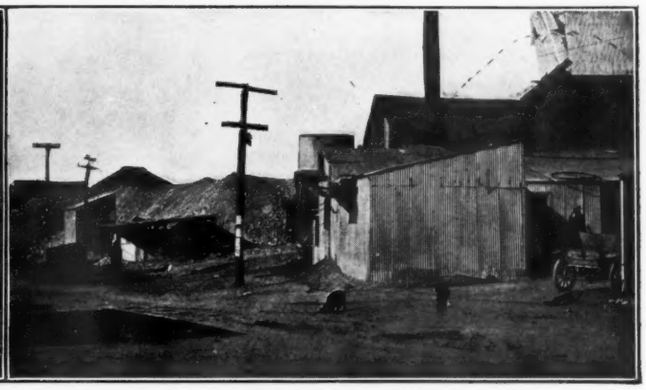


Fig. 2. Dotted Line Shows Flight of Receiver

At the time of the explosion the gage pressure was about at the usual point of 100 lb. The receiver was equipped with a safety valve in working order. The compressor was of the piston-inlet type, with a throttled outlet for unloading its low-pressure cylinder at 100 lb. pressure, thus the compressor "unloaded" automatically at about 4 lb. before the safety valve would open.

Not a single rivet had started nor did the seams show any indication of weakness. The grease and dust and noncorrosive paint over them looked exactly as it did before the explosion. One broken edge was almost in the middle of a plate, and continued straight around the cylinder, even where it crossed the lap seam. The other broken edge was about the same in appearance and 2 in. from the row of rivets in the same plate.

Fig. 3. Compressor Room after Explosion

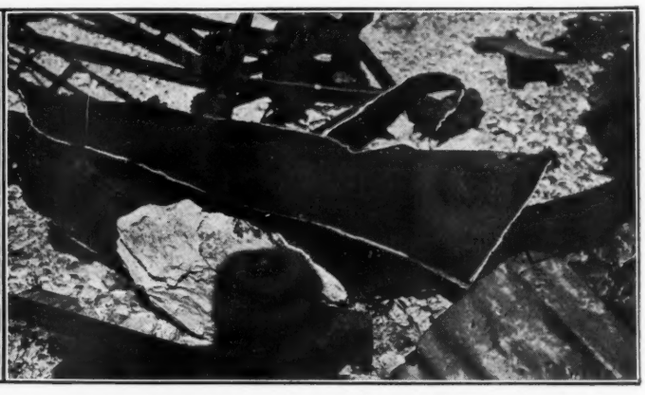


Fig. 4. Pieces of Plate Torn from Receiver

The rivets and seam were perfect, but the broken edges had been torn like a piece of cardboard (not square across but rough and laminated, tearing back ½ in. or more and splitting the sheet).

The 8-ft. end that had been blown away by the explosion was dry and burned and showed no sign of grease on its interior, nor did the pieces that had been torn out, but the 6-ft. end of the receiver that remained practically in its original position, had 2½ in. of grease or half solidified oil on the bottom, as it lay on its side, while the entire inside was lined with a scum of grease ¼ in. thick.

Apparently the grease and oil, which had accumulated for 7½ yr., had formed with the dust a scum on the interior area of the receiver which was too thick to flow to the drain valve. This had dried and baked upon the inside of the receiver. From the fact that the end that remained in place was still coated with grease and the pieces that were blown away by the explosion were devoid of grease and dry and clean, and appeared in spots like they might have been burned, it would appear that the vapor from this grease, held under pressure and the heat of compression, had become ignited, causing an explosion of lubricating-oil gas.

Many mines have air compressors and receivers that are not properly arranged for draining and there are probably other uses of compressed air in which defective conditions exist. It would seem that a manhole, with occasional cleaning and scraping of the interior of the receiver, should be provided as a means of preventing receiver explosions.

N. A. S. E. Educational Program

The educational program of the National Association of Stationary Engineers will include five lines of work, which will be begun as soon as practicable.

For lecture work among subordinate associations, lantern slides will be loaned by the national educational committee free of charge under certain conditions to be announced shortly. The subjects to be covered are steam; combustion and smoke prevention; steam turbines; boilers and superheaters; indicator diagrams, and gas engines. From 20 to 40 slides will be provided for each subject.

A catalog describing the pictures and giving information which will be helpful to anyone preparing lectures on the subjects in question will be sent to association secretaries as soon as the lists are complete. Three or four sets of slides will be ready for use by Jan. 1. Members having slides are requested to send duplicates to the Advisory Board, N. A. S. E., Box 81, Back Bay, Boston, Mass. In this way the board will have a large collection available.

The board during the coming year intends establishing a question department relating to plant operation and any technical work in which members may wish to be informed; these will be answered confidentially. The board is not anxious to waste its time over puzzles or trick questions, but will endeavor to be of practical assistance to members needing operating information.

As a means of encouraging educational work, subordinate associations will be offered prizes for holding the most instructive single meeting between Nov. 1 and May 1. To compete, an association must send a complete report of the meeting, signed by the president and secretary, stating the number of persons present and taking part in the discussion, and giving the discussion in full. The award will be based entirely upon these reports. Four prizes are offered of steam-engine indicator outfits of equal value, which will be awarded after May 1, 1913.

New Haven Meeting of A. S. M. E.

On Nov. 13 the New Haven, Conn., branch of the American Society of Mechanical Engineers held an afternoon and evening session in the lecture room of the Mason Laboratory of Mechanical Engineering. E. S. Cooley was chairman of the afternoon meeting and Prof. L. P. Breckenridge of the evening session.

Papers on the transportation of material in manufacturing plants were presented by C. F. Bennett, of the Stanley Works, New Britain, and by Herbert L. Seward, of the mechanical engineering department of the Sheffield Scientific School, Yale University. The second paper was illustrated by lantern slides. The papers were discussed by Arthur Brewer, superintendent of the Bridgeport Brass Co.; C. F. Hutchins, Sprague Electric Co., and Mr. Goldstein, of the Link-Belt Co. The remarks of the two last speakers were illustrated. C. W. Rice and Prof. Breckenridge also discussed the papers.

Written discussions were sent in by C. D. Rice, superintendent of the Underwood Typewriter Co., and Harwood Frost, of the Brown Portable Elevator Co.

The general subject for the evening session was "Industrial Education." Calvin W. Rice, secretary of the society, made a short address on the work of local branches of the American Society of Mechanical Engineers. This was followed by addresses and discussion on the general subject with especial reference to its present development in Connecticut. The work now being done by the state was described by F. J. Trinder, principal of the New Britain Trade School.

Both meetings were well attended and much interest in the subjects presented was manifested.

OBITUARY

Clement A. Griscom

On Nov. 10, Clement A. Griscom, one of the founders and the first president of the Society of Naval Architects and Marine Engineers, died at his home in Philadelphia, Penn.

Mr. Griscom was born in Philadelphia, Mar. 15, 1841, the son of Dr. John D. Griscom, a prominent Philadelphia physician. He received his early education in public and private schools. In 1857 he became a clerk for Peter Wright & Sons, shipping merchants, of Philadelphia. In 1863 he was admitted to partnership at the age of 22.

Mr. Griscom was one of the first in this country to recognize the growing ascendency of steam over sails for commercial purposes. He was elected president of the International Navigation Co. in 1888 and in 1902 was associated with J. P. Morgan in forming the International Mercantile Marine Co.

PERSONAL

W. R. Addicks, vice-president of the Consolidated Gas Co., of New York, has been elected president of the American Gas Institute.

R. D. Tomlinson has resigned his position as manager of the condenser and steam-engine department of the Allis-Chalmers Co., to become vice-president and general manager of the Grangers Lime & Marble Co., of Danbury, Conn. This will bring Mr. Tomlinson back to the scenes of his early life. Born and educated in Connecticut, he came to New York with the John F. Bushnell Co., became power-plant engineer for the Interborough Rapid Transit Co. in some of its earlier stages, leaving that position to become manager of the condensing department of the Allis-Chalmers Co., where he was later given the management of the steam-engine department also. He is a member of the American Society of Mechanical Engineers, past president of the National Association of Stationary Engineers, and one of the founders of the Institute of Operating Engineers, and if he carries into this new connection the same interest and intelligently applied endeavor which have marked his course as an engineer, the success which all his friends wish for him will be assured.

Twenty-four thousand men, as compared with 17,600 last year, are now employed by the Westinghouse companies. Their business is close to record levels.

Moments with the Ad. Editor

Here's a little problem in advertising for you.

Supposing a friend should ask your opinion of Sapolio, Elgin or Waltham Watches, Steinway Pianos, Yale Locks, Tiffany Jewelry, Royal Baking Powder, Pear's Soap, Quaker Oats, Eastman Kodaks, Victor Talking Machines or Uneeda Biscuits?

You may never have used any of them or even seen half of them, but the chances are 10 to 1 you'd say, "Why, sure, those articles are all right—you'll make no mistake in buying any of them," or words to that effect.

Now the question is, on what is your opinion based? That's the problem we're getting at.

And again it's 10 to 1 that your answer will be, "Well, I know all about them because I have read so many advertisements about them for so many years—and I know they are *good* because they have been so extensively advertised."

All of which reasoning is almost but not quite correct.

In reality you must turn your reason right around.

Those articles are not *good* because they are so well-known, but *they are well-known because they are so good.*

That's the *real* answer to the problem although the final result is the same to you, the buyer.

In other words, it's the quality and satisfaction-to-the-buyer in those goods that have made it highly profitable for the manufacturers to spend hundreds of thousands of dollars in advertising them to you.

For there is no quicker way on earth to get rid of a fortune than by advertising poor, unsatisfactory goods—

And there's no *better* investment than advertising *good, high quality products.*

Advertisers realize this and so do publishers—

And therein lies the basic reason why advertising protects your pocketbook.